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PERFORMANCE OF LUBRICANTS: OILS AND GREASES ON WEAR TESTS, COMP--ETC(U)
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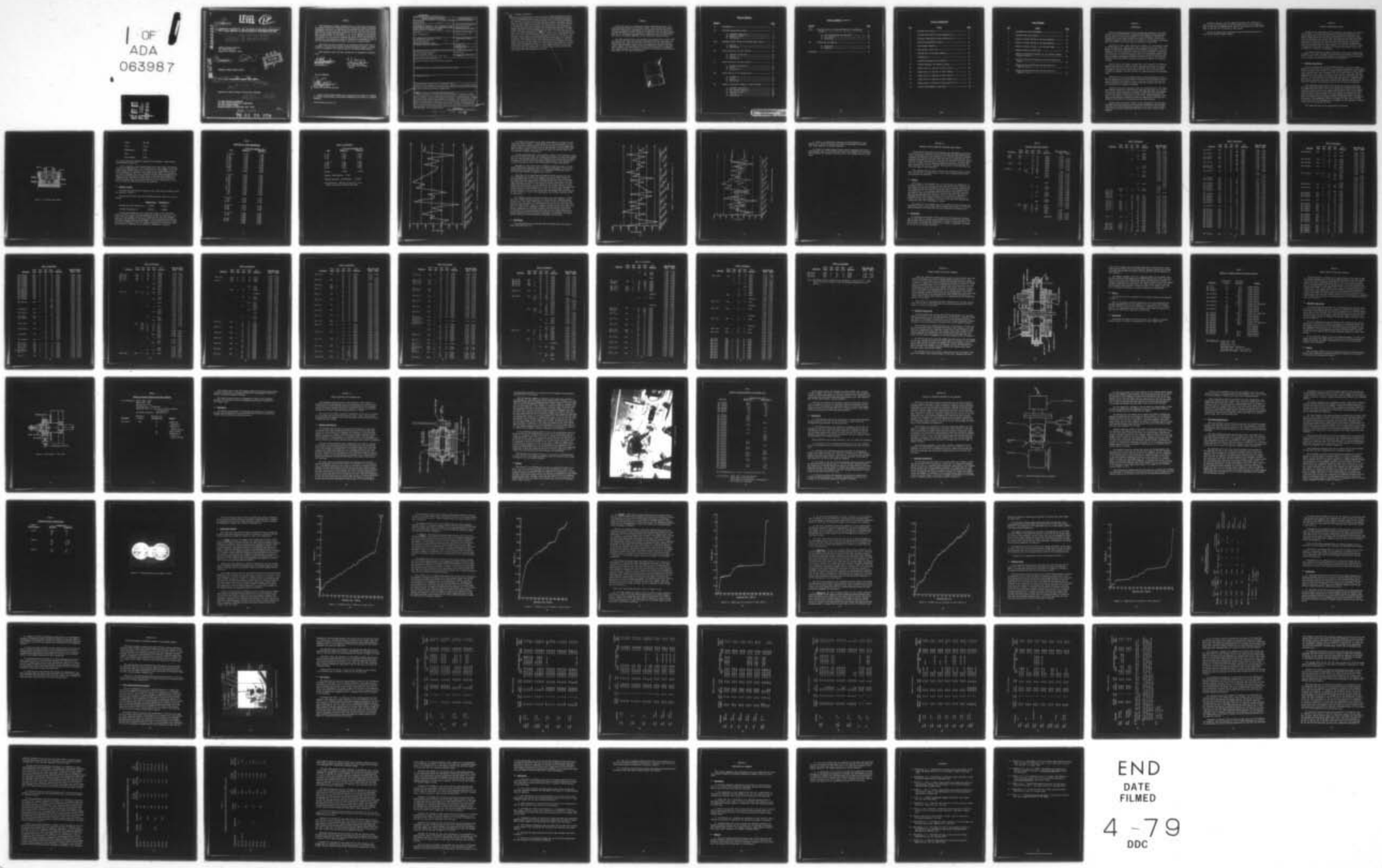

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6 PERFORMANCE OF LUBRICANTS: OILS AND GREASES ON WEAR TESTS, COMPACT MATERIALS IN BALL BEARINGS, AND SPUTTERED COATINGS ON GAS-BEARING COUPONS.

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TECHNICAL REPORT AFML-TR-78-126

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9 Final Report f[redacted]Jan[redacted]975 - Apr[redacted]978 2 14

15 F33615-75-C-5116

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1. REPORT NUMBER: AFML-TR-78-126	2. GOVT ACCESSION NO. / 3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle): PERFORMANCE OF LUBRICANTS: OILS AND GREASES IN WEAR TESTS, COMPACT MATERIALS IN BALL BEARINGS, AND SPUTTERED COATINGS ON GAS-BEARING COUPONS.	5. TYPE OF REPORT & PERIOD COVERED: Final Technical Report 1/2/75 to 4/14/78
	6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s): Karl R. Mecklenburg	8. CONTRACT OR GRANT NUMBER(s): F33615-75-C-5116
9. PERFORMING ORGANIZATION NAME AND ADDRESS: Midwest Research Institute, 425 Volker Boulevard, Kansas City, Missouri 64110	10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS: 7343
11. CONTROLLING OFFICE NAME AND ADDRESS	12. REPORT DATE: September 1978
	13. NUMBER OF PAGES: 93
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office): Air Force Materials Laboratory, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio 45433	15. SECURITY CLASS. (of this report): Unclassified
	15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Lubricant Compact Materials, Friction and Wear, Lubrication, Ball Bearing Operation, Wear-Lives, Solid Lubrication, Self-Lubricating Materials, Lubricants, Oils, Greases, Four-Ball Wear Tests, Wear Tests, Spindle Tests

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This Final Report for a 39.5-month contract contains the results of several projects conducted on the program. The first project was a continuation of a long-term repeatability study on the Four-Ball wear testers. The work was done to assure that the two test machines were constant in performance. The data confirm that the machine performance was as constant as could be statistically expected, both from machine to machine and for each machine as a function of time. The second project was to conduct Four-Ball wear tests on various oils and greases to determine the over

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20. ABSTRACT (Concluded)

ability of the lubricants to prevent wear at various conditions of speed, load, and temperature. Data from 1,111 tests on 200 lubricating materials are reported. The third and fourth projects were to determine performance lives of various greases on the Pope and Navy spindles. Equipment descriptions and test results are included. The fifth project was grease testing in the Sikorsky rigs. These rigs are described and the test results presented. The sixth project discussed was the long-term testing of lubricant compact materials in ball bearings. Over 31,000 hr of operation at 1,790 rpm in a vacuum environment were logged for each of the five bearings (size 204). Two of the five bearings were still operating at 54,000 and 94,024 hr. The other three failed, with operating times of 31,075 hr, 32,984 hr, and 52,000 hr. Wear rates were determined and predictions made as to expected wear-lives. The seventh project discussed was the evaluation of 163 sputtered coatings of various materials, using different thicknesses and techniques. The coatings were applied to small coupons and tested for friction coefficients and wear-lives. Most of these tests revealed that much more work must be done by the supplier to provide lower friction coefficients and consistently long operating wear-lives.

FOREWORD

This Final Report was submitted by Midwest Research Institute, 425 Volker Boulevard, Kansas City, Missouri 64110, under Contract No. F33615-75-C-5116, "Exploratory Development on New Lubricants and the Effects of Extreme Environments on Their Behavior," Project No. 7343, Task No. 734303, with the Air Force Materials Laboratory, Wright-Patterson Air Force Base, Ohio. Mr. F. C. Brooks, AFML/MBT, was the laboratory project monitor.

The work reported herein was conducted from January 1975 to April 1978, and includes data from work started in September 1967. The manuscript was submitted by the author in June 1978. Other reports on the same contract include AFML-TR-76-240, "High Pressure and Temperature Effects on the Viscosity, Density, and Bulk Modulus of Two Liquid Lubricants," and AFML-TR-78-5, "High Pressure and Temperature Effects on the Viscosity, Density, and Bulk Modulus of Four Liquid Lubricants."

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SECTION I

INTRODUCTION

This is the Final Report for the research work conducted on Air Force Contract No. F33615-75-C-5116, "Exploratory Development on New Lubricants and the Effects of Extreme Environments on Their Behavior." Sections II through VIII of this report present the results of several projects that have been conducted during the past 39.5 months of the contract. Conclusions drawn from results of all the projects are given in Section IX.

Section II of this report contains the results of a repeatability study on the Four-Ball wear testers. This work, a continuation of the effort reported in Contract No. AFML-TR-75-32, "Performance of Lubricants: Oils and Greases in Wear Tests and Compact Materials in Ball Bearings," issued May 1975 (Ref. 1), was performed to assure that the test machines remained relatively constant in their performance during an extended period. Data show that the machine performance remained as constant as can be statistically expected, both from machine to machine and for each machine as a function of time.

Section III of this report contains data on the ability of various lubricants and hydraulic fluids to prevent wear under different test conditions. These data were collected with a Four-Ball wear tester to augment and check on results from other sources; the work was not intended to be a definitive program in itself. The data are presented herein for reference only.

Sections IV, V, and VI contain results on the performance of various greases operating on the Pope Spindles (Section IV), Navy Spindles (Section V), and Sikorsky rigs (Section VI). As in Section III, the data were collected to complete and check on results from other sources; the work was not intended to be a definitive program.

Section VII contains additional wear rate determinations of the lubricant compact separators in size 204 ball bearings. This work was a continuation of the effort reported in Contract No. AFML-TR-74-181, "Performance of Lubricant Compact Materials in Ball Bearings," issued September 1974 (Ref. 2) and updated in Reference 1. Three of the compact-lubricated bearings failed during this contract period. The remaining two bearings have 54,818 hr and 84,024 hr of operation, and are still running.

Section VIII contains the results of the evaluation of 163 sputtered coatings of various materials, using different thicknesses. The coatings were applied to small gas-bearing coupons and tested for friction coefficients and wear-lives. Additional work needs to be done on this coating process.

Viscosity, density, and bulk modulus properties were determined at high pressures and temperature on this program. Results of this work are given in Contract No. AFML-TR-76-240 (Ref. 3) and Contract No. AFML-TR-78-5 (Ref. 4) and will not be repeated in this report.

Section IX contains the conclusions which were derived from the results of the various program studies.

SECTION II

FOUR-BALL REPEATABILITY STUDY

A study of the repeatability of the Four-Ball wear testers was started in 1972 (Ref. 1) and continued through this contract. Repeatability was questioned because of some "unusual" observations of the wear scar data. Using a single lubricant with the same test conditions produced wear scars that seem reversed, i.e., larger scars were observed with fluids containing anti-wear additives than with the same fluids without the additive. There were also instances in which one of the two machines produced scars consistently greater than the other machine, but only for a period of time. A program was initiated to determine if the wear scar results were in any way related to using the two machines.

The program consisted of running identical tests, using a standardized lubricant, at periodic intervals. In this manner, one machine could be compared to itself and to the other machine as a function of time. The results of the program are presented in this part of the report.

A. Equipment Description

The test configuration of the Four-Ball machine is shown in Figure 1. Basically, three balls are clamped in a triangular arrangement, and the fourth ball is rotated in the "pocket" or triangle formed between the three clamped balls. Rotation speed can be changed, with the speeds of 600 and 1,200 rpm being used the most. Loads are varied to suit the test conditions. The most commonly used loads between the rotating ball and the three stationary balls are 1, 4, 10, 20, and 40 kg, with 75 kg used occasionally. A number of ball materials are available, but 52100 and M-10 tool steel are most generally used. Test duration and temperature are two more of the test variables.

The operating sequence starts with heating the ball pot (containing the three stationary lower balls) to the desired test temperature. After the temperature stabilizes, the load is applied and rotation of the spindle started. A timer, preset to the correct test duration, turns off the drive motor and the electrical heaters. The test operator then removes the load. After cool-down, the test balls are removed, chemically washed, and wear scars on the three lower (stationary) balls are measured with a tool-maker's microscope. Two measurements, taken at 90 degrees to each other, are made on each ball. All six measurements are averaged, and this average is reported as the wear scar value for the test.

The conditions used in the repeatability study were:

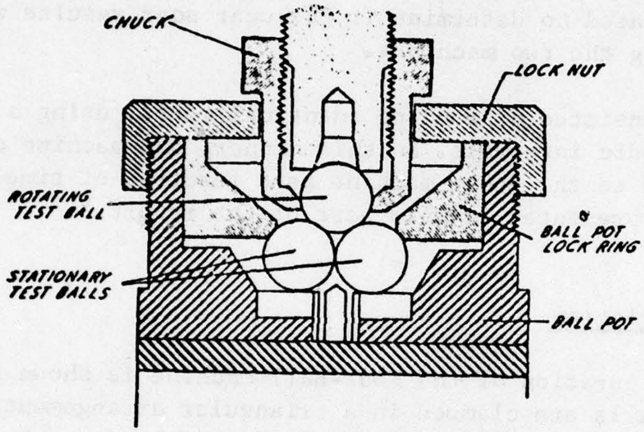

Figure 1 - Four-Ball Wear Tester

Speed	600 rpm
Load	10 kg
Temperature	167°F
Time	1 hr
Ball material	52100

All of the tests were conducted using ELO 67-22 (KG-80), a super-refined paraffin-base mineral oil.

The two machines, those normally used in the daily work, were identified only as Machines Nos. 2 and 3. Both machines were assigned their individual spindle chucks and ball pots. There was no interchanging of parts between machines. In addition, each tapered spindle chuck was tailored to its spindle housing. After total indicated run-out (TIR) measurements were made, the relative position between chuck and housing that produced the minimum TIR was identified by punch marks on both the spindle chuck and spindle housing. Thus, the chuck was returned to the same position at the start of each test. The TIR was less than 0.002 mm.

B. Results of Study

The wear scar data from the complete study (going back to February 1972) are presented in Table 1.

Working with the data from each individual machine results in the following:

	Machine No. 2	Machine No. 3
Average wear scar diameter, mm	0.23505	0.24678
Standard deviation, mm	0.01338	0.00988

The data for Machine No. 2 are plotted in Figure 2. The individual values are almost equally divided between those above the average (21) and those below the average value (19) of 0.25305 mm. The one- and two-sigma bands are also indicated on the graph. These values represent multiples of the sample standard deviation both added to and subtracted from the average value. (See References 5 and 6 for further information on statistical analysis and how the technique can be applied to experimental testing.)

TABLE 1

REPEATABILITY STUDY MEASUREMENTS

Date	Wear Scar Diameter, mm Machine 2	Machine 3
17 Feb 72	0.239	0.262
22 Mar	0.239	0.246
11 Apr	0.231	0.224
25 May	0.254	0.244
26 Jun	0.249	0.251
31 Jul	0.264	0.251
31 Aug	0.254	0.239
29 Sep	0.239	0.267
1 Nov	0.251	0.259
4 Dec	0.241	0.246
5 Jan 73	0.218	0.257
1 Feb	0.274	0.239
26 Feb	0.267	0.259
30 Mar	0.251	0.244
1 May	0.249	0.244
1 Jun	0.257	0.231
29 Jun	0.234	0.257
20 Jul	0.249	0.231
4 Sep	0.272	0.239
5 Oct	0.267	0.241
1 Nov	0.262	0.246
22 Jan 74	0.277	0.251
8 Apr	0.272	0.241
12 Jul	0.267	0.257
23 Jan 75	0.264	0.229
18 Apr	0.257	0.254
24 Sep	0.254	0.246
15 Jan 76	0.249	0.262
19 Feb	0.246	0.249
	0.246	0.249
20 Feb	0.262	0.241
23 Feb	0.249	0.249
	0.244	0.239

Table 1 (continued)

Date	Wear Scar Diameter, mm Machine 2	Machine 3
15 Jun	0.259	0.254
12 Oct	0.244	0.249
15 Dec	0.226	0.239
3 Feb 77	0.257	0.262
12 May	0.264	0.236
14 Sep	0.262	0.246
12 Dec	0.262	0.241
Average:	0.2530	0.2468

Average, Both Machines: 0.2499

Standard Deviation, Both Machines: 0.012103

Test Conditions: 600 rpm; 10 kg; 75°C; 1 hr; 52100 steel balls; ELO 67-22 lubricant.

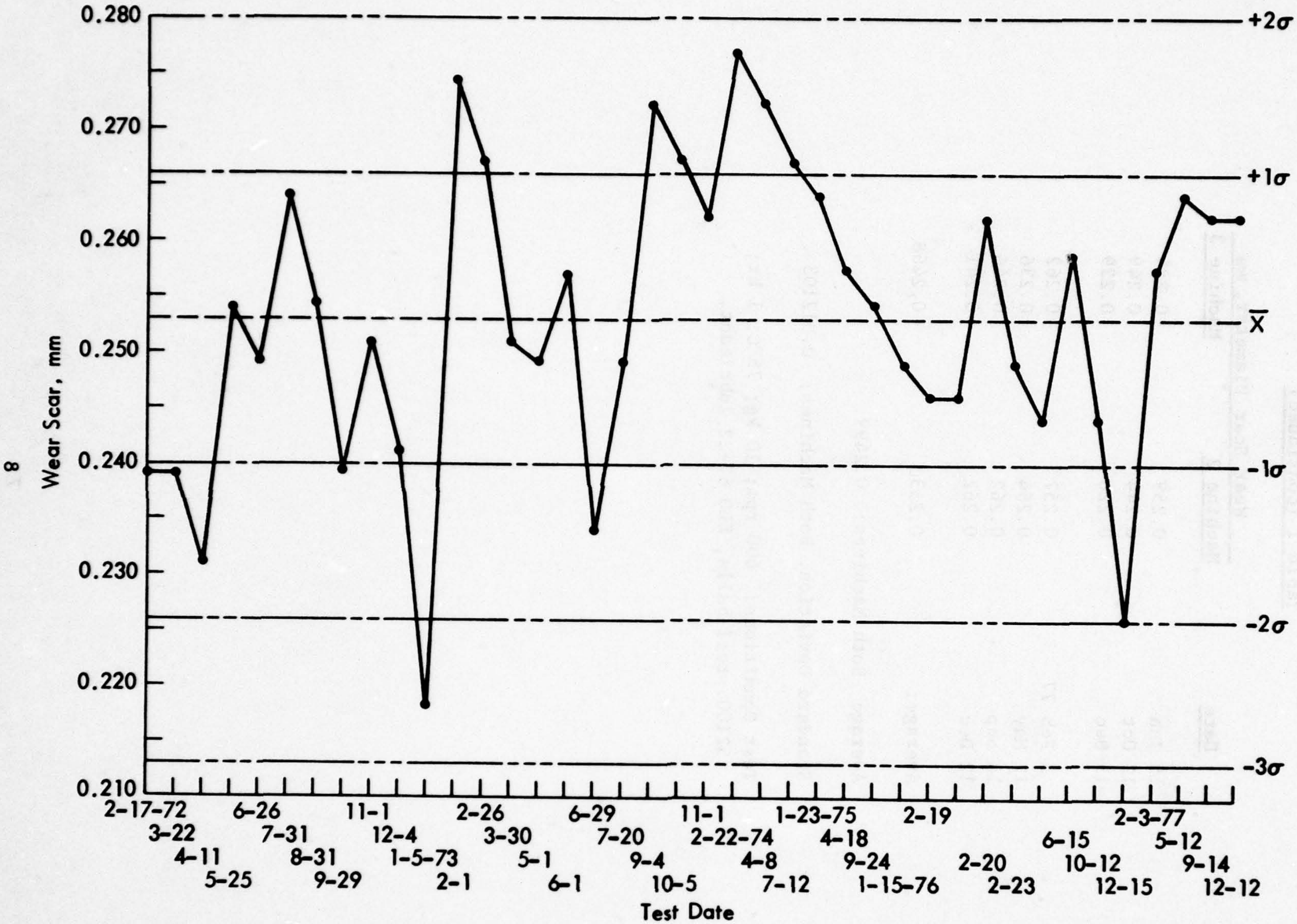

Figure 2 - Four-Ball Repeatability Study, Machine No. 2

For normal distribution of the data points about the average, the one-sigma band should contain at least 68.26% of the 40 data points, or 27.30 points. The band actually contains 26 of the 40 points. The two-sigma band should contain at least 95.45% of the 40 data points, or 38.18 points. This band actually contains 38 of the 40 points. Thus, the data do seem to fit the criteria of normal distribution.

The data for Machine No. 3 are plotted in Figure 3. The individual values show that 10 points are above the average and 22 points are below the average. The one- and two-sigma bands are also indicated on the graph, with the bands containing 26 and 38 of the data points, respectively, as with Machine No. 2. Again, the data do seem to fit the criteria of normal distribution.

Comparison of data from these two machines would show there is a difference between the machines, with a statistical probability of 21.3%. The repeatability data show that there is no significant difference between the wear scars produced on Machines Nos. 2 and 3.

Since there is no significant difference between machines, the data can be combined and plotted as in Figure 4. As can be seen in Figure 4, the individual values are equally divided between those above the average and those below the average value of 0.2499 mm. The one-, two-, and three-sigma bands are also shown on the graph. For normal distribution, the one-sigma band should contain at least 68.26% of the 80 data points, or 54.61 points. The band actually contains 60 of the data points. The two-sigma band should contain at least 95.45% of the points, or 76.36 points. The band actually contains 77 of the data points. The three-sigma band should contain at least 99.73%, or 79.78 of the data points, and in fact, contains all 80 of the points.

The average value for the wear scar data from both machines is 0.2499 mm, with a sample standard deviation of 0.01210 mm. For a maximum spread of data of 2σ (two times 0.01210 mm), any value of wear scar diameter from 0.226 to 0.274 mm would be an acceptable value. Any value of wear scar outside that band would be subject to question, especially any value over 0.274 mm. If a repeat of the experiment produced the same results, the machine would be subjected to a detailed inspection. The most likely fault would be failure of the spindle bearings. For work done on previous efforts, the spindle bearings have failed and were replaced. The possibility of spindle bearing failure is often overlooked in Four-Ball wear testing.

C. Conclusions

Two conclusions can be drawn from this continuing study, the same as drawn previously (Ref. 1).

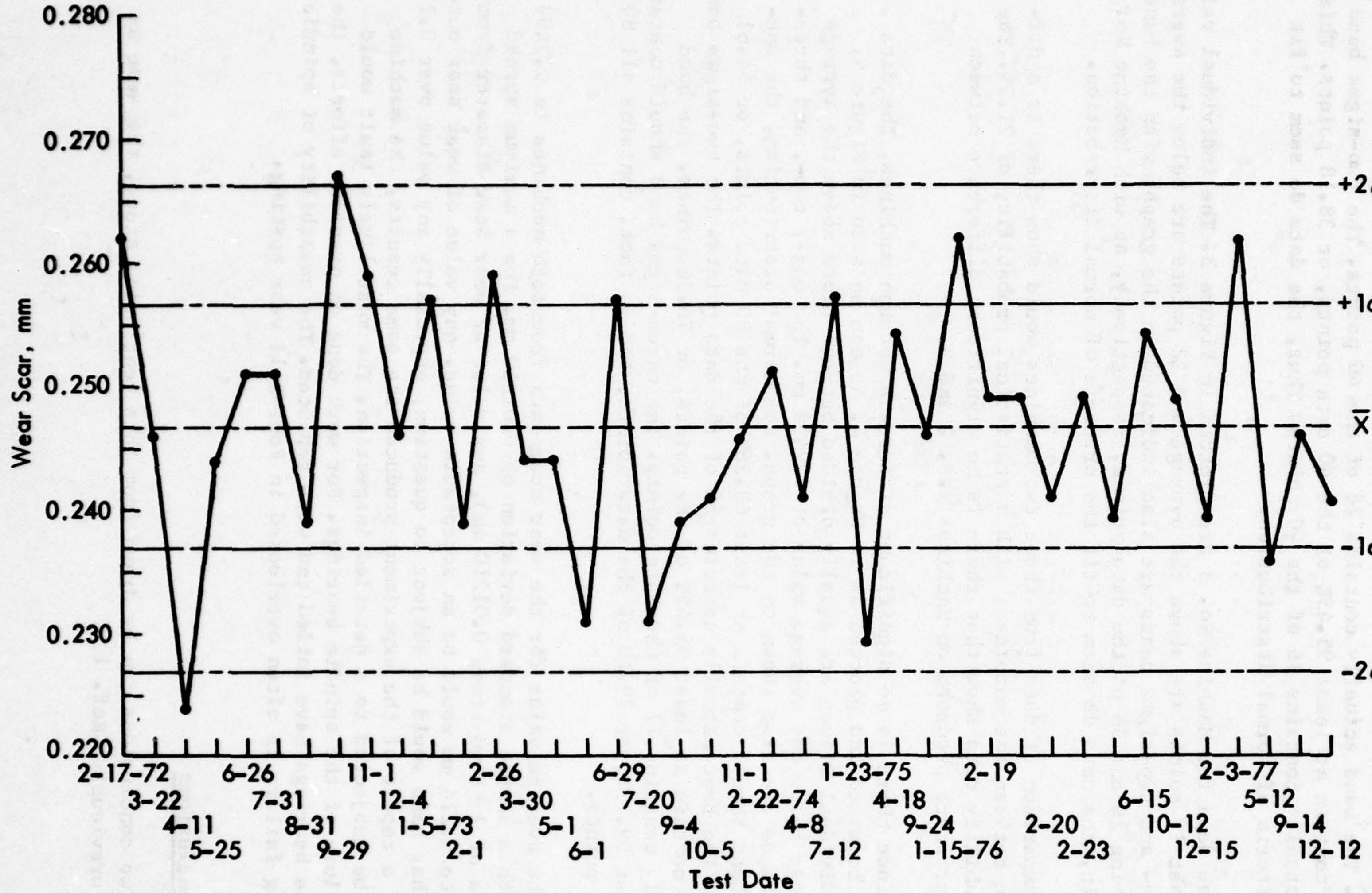

Figure 3 - Four-Ball Repeatability Study, Machine No. 3

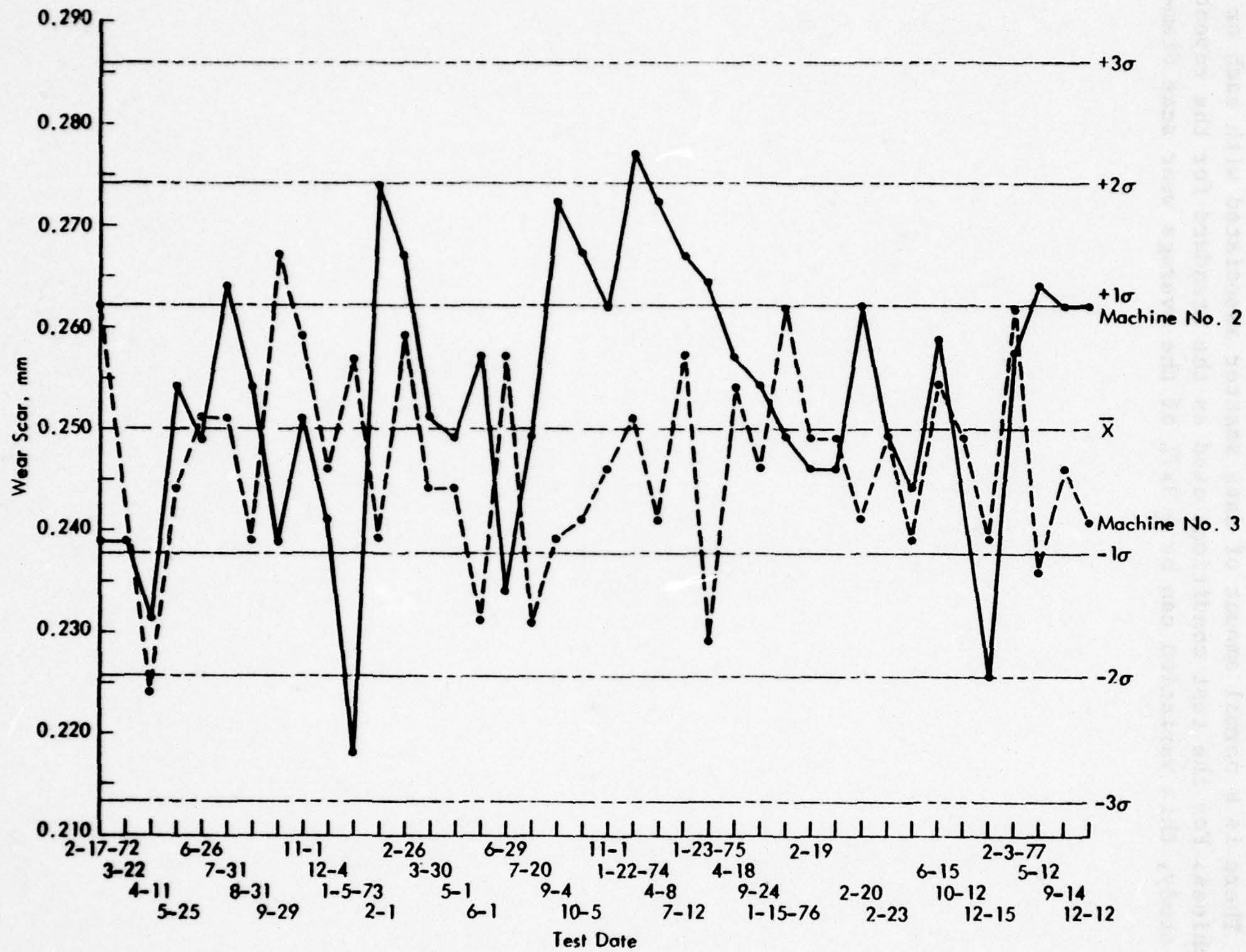

Figure 4 - Four-Ball Repeatability Study

1. There is no significant difference between Machine No. 2 and Machine No. 3 as far as the wear scar data are concerned. The results from either machine are equally acceptable.

2. There is a normal amount of data scatter associated with each or both machines. For the test conditions used as the standard for the repeatability study, this variation can be $\pm$ 9.7% of the average wear scar diameter.

SECTION III

LUBRICANT RESULTS USING THE FOUR-BALL WEAR TESTERS

In the normal operation on this program, wear performance data on various lubricants and hydraulic fluids are required. The data required most often are Four-Ball wear scar information on a specific material. Generally speaking, the Four-Ball wear scar tests are used as a screening test for potential lubricants. Quite often, the tests are conducted to verify results produced by other laboratories, to complete the data for a specific lubricant, or to collect data at different operating conditions. As such, the data presented are not for a specific study on one or a few materials, but for a few tests on many materials.

The equipment used for these tests has been described in Section II-B. The tests were run on Machines Nos. 2 and 3, with an approximately equal split between the machines.

A. Results

The purpose of any Four-Ball test is to determine the "lubricity" of a material. That is, the smaller the wear scars on the three clamped balls, the better the lubricating capability of the tested material, provided the test conditions are the same. The basic operation of the Four-Ball tester is intended to be in the boundary lubrication regime; although after the wear scar has been generated and/or if the loads are small, the operation may become hydrodynamic. The Four-Ball tester is used as a lubricant-screening device. The accepted criterion for lubricant performance is a "small" wear scar when compared to the size of the wear scar produced with a lubricant whose fluid performance is known.

The results of the Four-Ball wear tests conducted on this contract are presented in Table 2. This lengthy table contains the results of 1,111 tests which encompasses 200 fluids. The data are presented for reference only.

B. Conclusions

No conclusions were drawn from the wear scar data presented. Information on such things as chemistry and formulation was lacking for each fluid, and the specified testing was insufficient to permit a good analysis of the data. However, these data are expected to be useful to engineers interested in any one of the specific materials.

TABLE 2

FOUR-BALL WEAR TEST RESULTS

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
6195	600	1	10	75	52100	0.267	
7558	600	1	1	75	52100	0.175	0.168
			10	75	52100	0.269	0.249
			40	75	52100	0.625	0.574
7558X	600	1	1	75	52100	0.165	0.152
			10	75	52100	0.257	0.239
			40	75	52100	0.457	0.450
EL067-22	600	0.17	75	200	440-C		3.820
		0.22	75	200	440-C	4.008	
		2	10	75	440-C	0.254	0.246
					52100	0.246	0.236
				200	440-C	0.470	0.549
						0.417	0.452
						0.483	0.521
					52100	0.391	0.470
						0.323	0.345
			40	75	440-C	1.369	1.473
					52100	0.587	0.549
						0.627	0.607
				200	440-C	2.802	2.800
					52100	0.503	0.498
			75	75	440-C	3.434	4.900
						3.917	5.395
					52100	0.902	0.831
				200	52100	1.367	1.735
						1.496	1.849
	1200	0.08	75	75	440-C	8.524	6.007
		0.12	40	40	440-C		4.183
		0.2	75	200	52100	5.245	
		1	40	75	440-C	3.934	
					52100	0.544	0.546
						0.518	0.554
					52100/TCP	0.495	0.511
						0.607	0.518

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Wear Scar (mm) Rig 3
		2	10	75	440-C	0.587	0.249
						0.234	0.244
					52100	0.259	0.274
				200	440-C	1.504	1.461
					52100	0.376	0.391
			40	75	440-C	5.433	2.060
						4.514	2.492
						4.656	2.502
						4.453	5.128
					52100	0.584	0.572
				200	440-C	5.519	4.288
						4.371	3.622
					52100	1.933	1.681
						0.810	1.085
			75	75	440-C	6.038	5.857
					52100	3.404	2.365
						3.028	2.403
						0.968	2.291
							2.347
				200	52100		3.063
						5.118	5.217
						4.671	4.935
	1800	1	1	75	440-C	0.198	
ELO 67-22	600	1	10	75	52100	0.691	
+Corrosion						0.765	0.691
Inhibitor							
ELO 67-22	600	2	10	200	52100	0.447	0.333
Degassed			40	200	52100	0.815	0.772
			75	200	52100	1.506	1.760
LJG-245 A	1200	1	40	75	52100	0.848	0.815
LJG-245 B	1200	1	40	75	52100	0.879	0.859
LJG-247 A	1200	1	40	75	52100	0.546	0.574
LJG-247 B	1200	1	40	75	52100	0.719	0.716
LJG-247 C	1200	1	40	75	52100	0.675	0.759
LJG- A 4	600	2	10	75	52100	0.460	0.518
				204	M-10	1.179	1.060
			40	75	52100	0.737	0.744
				204	M-10	1.580	1.643
LJG- A 5	600	2	10	75	52100	0.277	0.264
				204	M-10	0.262	0.257
			40	75	52100	0.495	0.513
				204	M-10	0.427	0.427
MCG 68-63	1200	2	40	75	M-50	0.912	0.886
MCG 70-27	1200	2	40	75	52100	0.879	0.884
MCG 70-28	1200	2	40	75	52100	0.681	0.996
MCG 7316322	1200	2	40	75	52100	1.156	1.143
		4	40	AMB	52100	1.389	1.328
						1.488	1.438

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
MCG 731623	1200	2	40	75	52100	1.298	1.163
		4	40	AMB	52100	1.405	1.443
MCG 7316324	1200	2	40	75	52100	1.173	1.105
		4	40	AMB	52100	1.270	1.293
MCG 7319925	1200	2	40	232	M-50	1.054	0.963
MCG 7325431	1200	2	40	75	52100	1.240	0.953
		4	40	AMB	52100	1.341	1.255
MCG 7325432	1200	2	40	75	52100	1.052	1.143
		2	40	AMB	52100	1.364	1.191
MCG 74171026	1200	2	75	75	M-10	0.729	0.800
MCG 74353056	1200	2	40	75	52100	1.290	1.153
		4	40	AMB	52100	1.361	1.290
MCG 75050001	1200	2	40	75	440-C		2.995
					52100	0.884	
					M-10	0.384	0.371
					M-50		0.330
MCG 75218025	1200	2	40	75	52100	1.511	2.123
MCG 75245026	600	2	40	75	52100	0.777	0.767
	1200	4	40	AMB	52100	1.072	1.262
MCG 75283029	1200	2	40	75	52100	1.915	2.040
MCG 75283030	1200	2	40	75	52100	2.086	2.252
MCG 75283031	1200	2	40	75	52100	1.890	1.689
MCG 75283032	1200	2	40	75	52100	1.806	1.890
MCG 75288033	1200	2	40	75	52100	1.242	1.184
		4	40	AMB	52100	0.958	0.991
MCG 76019013	1200	2	40	75	52100	1.095	1.229
MCG 76019014	1200	2	40	75	52100	1.361	2.060
MCG 76050015	1200	1	75	75	52100	1.095	0.942
MCG 76096015	1200	2	40	75	52100	1.140	1.260
MCG 76096016	1200	4	40	AMB	52100	1.410	1.351
MCG 76096017	1200	2	40	75	52100	1.234	2.029
				232	M-10	1.034	0.953
MCG 76104019	1200	2	40	75	52100	0.605	0.589
			75	75	52100	0.864	0.866
MCG 76132022	1200	2	40	75	52100	1.483	2.164
MCG 76132023	1200	2	40	75	52100	1.557	2.192
MCG 76132024	1200	2	40	75	52100	1.666	1.829
MCG 76132025	1200	2	40	75	52100	1.514	2.162
MCG 76132026	1200	2	40	75	52100	2.065	1.593
MCG 76132027	1200	2	40	75	52100	1.641	1.773
MCG 76132028	1200	2	40	75	52100	1.621	2.108
MCG 76132029	1200	2	40	75	52100	1.494	1.542
MCG 76140031	1200	2	40	75	52100	1.542	1.877
MCG 76140032	1200	2	40	75	52100	1.034	2.116
MCG 76189034	1200	2	40	75	52100	1.468	1.994
				232	440-C	1.361	1.516
					M-10	1.209	0.813
					M-50	0.884	0.757
MCG 76212035	1200	2	40	75	52100	1.196	1.407

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
MCG 76235039	1200	2	40	75	52100	0.942	0.960
						0.983	1.115
		4	40	AMB	52100	1.265	1.344
						1.115	1.143
MCG 76236041	1200	2	40	75	52100	1.298	1.336
		4	40	75	52100	1.346	1.499
MCG 76267042	1200	2	40	75	52100	1.687	1.148
				232	M-10	0.853	0.818
MCG 76312003	1200	2	40	75	52100	0.709	0.983
				204	440-C	5.110	3.551
					M-10	2.558	1.869
					M-50	2.987	2.441
MCG 76313004	1200	0.17	40	204	440-C	4.663	
		1	40	204	440-C		4.580
		2	40	75	52100	0.866	0.897
				204	M-10	1.996	1.019
						2.426	1.077
					M-50	2.791	3.205
MCG 76313044	1200	2	40	75	52100	0.653	0.493
		4	40	AMB	52100	0.438	0.483
MCG 76313045	1200	2	40	75	52100	0.833	0.734
						0.841	0.785
		4	40	AMB	52100	0.744	0.686
MCG 76321057	1200	2	40	75	52100	0.726	0.770
MCG 76321058	1200	2	40	75	52100	0.828	0.820
MCG 76342060	1200	2	40	75	52100	1.905	2.169
				232	M-10	0.894	0.917
MCG 76342061	1200	2	40	75	52100	1.468	1.466
				232	M-10	0.977	0.861
MCG 77020318	1200	2	40	75	52100	1.044	0.658
MCG 77020319	1200	2	40	75	52100	1.671	1.341
MCG 77020320	1200	2	40	75	52100	1.521	2.433
MCG 77020321	1200	2	40	75	52100	1.387	1.473
MCG 77020822	1200	2	40	75	52100	0.912	0.836
			75	75	52100	1.003	0.820
MCG 77030326	1200	2	40	75	52100	0.782	1.549
MCG 77030327	1200	2	40	75	52100	0.968	1.003
MCG 77030328	1200	2	40	75	52100	2.106	2.855
MCG 77030329	1200	2	40	75	52100	1.773	2.408
MCG 77031432	1200	2	40	75	52100	2.200	2.167
				232	M-10	0.879	0.894
MCG 77052737	1200	2	75	75	52100	1.179	1.148
MCG 77052738	1200	2	75	75	52100	4.602	5.326
MCG 77081663	1200	2	40	75	52100	2.164	2.324
MCG 77081664	1200	2	40	75	52100	1.029	1.057
MCG 77083165	1200	2	40	75	52100	0.752	0.726

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
MCG 77083166	1200	2	40	75	52100	0.803	0.706
MCG 77110878	1200	2	40	75	52100	2.311	2.083
MCG 77110879	1200	2	40	75	52100	1.897	1.768
MCG 77110880	1200	2	40	75	52100	1.168	1.732
MCG 77110881	1200	2	40	75	52100	2.055	2.306
MCG 77110882	1200	2	40	75	52100	0.848	0.775
MCG 77111783	1200	2	40	75	52100	1.021	1.321
MCG 77111784	1200	2	40	75	52100	1.420	1.328
MCG 78010401	1200	2	40	75	52100	1.283	1.346
MCG 78010402	1200	2	40	75	52100	0.762	0.823
MCG 78010403	1200	2	40	75	52100	0.757	0.734
MCG 78010404	1200	2	40	75	52100	0.798	0.785
MCG 78010405	1200	2	40	75	52100	0.792	0.757
MCG 78010406	1200	2	40	75	52100	0.815	0.897
		4	40	AMB	52100	0.790	0.853
MCG 78012407	1200	2	40	75	52100	0.724	0.831
			75	75	52100	0.851	1.054
		4	40	AMB	52100	0.826	0.856
			75	AMB	52100	1.196	1.367
MCG 78032033	1200	2	40	75	52100	0.683	0.693
			75	75	52100	1.001	1.034
MIL-G-823261	1200	2	40	75	52100	1.433	2.240
				232	M-10	1.046	0.960
MIL-H-5606	1200	1	40	75	52100	1.031	0.996
MIL-H-5606-A	1200	1	1	75	52100	0.201	0.218
			10	75	52100	0.318	0.318
			40	75	52100	1.003	0.922
MIL-H-5606-B	1200	1	1	75	52100	0.214	0.211
			10	75	52100	0.305	0.302
			40	75	52100	1.052	0.823
MIL-H-5606-C	1200	1	1	75	52100	0.191	0.203
			10	75	52100	0.333	0.318
			40	75	52100	0.739	0.866
MIL-L-6085	600	1	1	75	52100	0.295	0.216
			10	75	52100	0.411	0.396
			40	75	52100	0.643	0.711
MIL-L-7808-W	1200	1	40	75	52100	0.879	0.462
						0.485	0.526
MLO 56-625	1200	1	40	75	52100	1.483	0.986
MLO 56-1050	1200	1	40	75	52100	1.540	1.448
MLO 57-104	1200	1	40	75	52100	1.575	0.947
MLO 69-35	600	2	10	75	52100	0.706	0.653
MLO 69-35+2%A2+5	600	2	10	75	52100	0.290	0.287
MLO 71-1	600	1	1	75	52100	0.175	0.216
			10	75	52100	0.290	0.302
MLO 71-2	600	1	1	75	52100	0.229	0.170
			10	75	52100	0.318	0.381

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
			40	75	52100	0.450	0.447
MLO 71-30	600	1	40	75	52100	0.599	0.592
MLO 73-65	600	1	40	75	52100	0.719	0.729
MLO 73-67	600	1	40	75	52100	0.632	0.610
MLO 73-91	600	2	10	75	440-C	0.772	0.765
					52100	0.569	0.569
				200	440-C	0.983	0.973
					52100	0.678	0.691
			40	75	440-C	1.059	1.016
				200	440-C	1.443	1.427
					52100	0.978	1.021
MLO 73-92	600	1.25	75	200	440-C	6.886	6.605
		2	10	75	440-C	0.269	0.254
						0.272	0.295
					52100	0.602	0.602
				200	440-C	0.518	0.663
						0.564	0.625
					52100	0.378	0.356
						0.363	0.439
			40	75	440-C	0.615	1.043
						0.955	0.993
					52100	0.874	0.937
				200	440-C	2.454	2.416
					52100	1.016	1.455
						0.998	1.257
			75	75	440-C	3.051	2.789
					52100	1.181	1.163
				200	52100	1.405	1.659
						1.311	1.339
	1200	0.08	75	75	440-C	4.872	5.499
		0.10	75	75	440-C	5.425	5.428
		0.47	40	200	440-C	5.403	
		0.50	40	75	440-C	5.132	
		2	10	75	440-C	0.381	0.386
					52100	0.620	0.691
				200	440-C	0.488	0.607
					52100	0.620	0.353
			40	75	440-C		1.887
						4.381	4.255
					52100	1.054	1.024
				200	440-C	5.199	3.018
							2.258
						5.027	1.458
					52100	1.900	1.882
			75	75	52100	2.512	2.096
						2.120	2.207
MLO 73-92 a	1200	1	40	75	52100	0.788	0.775
						0.767	0.772
						0.747	0.632

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Wear Scar (mm) Rig 3
						0.729	0.772
MLO 73-95	600	2	40	75	52100	0.747	0.838
	1200	2	40	75	52100	0.765	0.724
MLO 74-9	600	2	75	75	52100	0.594	1.001
						1.742	1.905
						1.026	1.013
				200	440-C	1.760	1.681
					52100	1.102	1.105
	1200	1	40	75	52100	0.564	0.455
						0.589	0.564
					52100/TCP	0.544	0.546
						0.549	0.541
		2	10	75	440-C	0.965	0.914
					52100	0.716	0.655
						0.716	0.615
				200	440-C	1.316	1.199
					52100	0.676	1.026
			40	75	440-C	2.366	2.131
					52100	0.742	0.668
				200	440-C	2.179	1.923
						2.487	2.309
					52100	1.057	1.270
						1.156	1.633
			75	75	440-C	2.474	2.433
					52100	0.980	1.618
						1.021	1.595
MLO 74-46	600	1	1	75	52100	0.163	0.165
			10	75	52100	0.257	0.249
			40	75	52100	0.505	0.498
MLO 74-47	600	1	1	75	52100	0.180	0.140
			10	75	52100	0.249	0.251
			40	75	52100	0.452	0.457
MLO 74-48	600	1	1	75	52100	0.175	0.206
			10	75	52100	0.267	0.244
			40	75	52100	0.584	0.589
MLO 74-49	600	1	1	75	52100	0.165	0.160
			10	75	52100	0.274	0.257
			40	75	52100	0.376	0.386
MLO 75-50	600	1	1	75	52100	0.145	0.180
			10	75	52100	0.284	0.277
			40	75	52100	0.414	0.439
MLO 74-51	600	1	1	75	52100	0.180	0.168
			10	75	52100	0.277	0.244
			40	75	52100	0.376	0.394
ML) 74-52	600	1	1	75	52100	0.213	0.233
			10	75	52100	0.221	0.231
			40	75	52100	0.470	0.373

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
MLO 74-67	1200	1	10	75	52100	0.376	0.381
			20	75	52100	0.511	
			40	75	52100	0.798	0.635
MLO 74-69	1200	1	10	75	52100	0.417	0.401
			20	75	52100		0.447
			40	75	52100	0.754	0.625
MLO 74-81	600	2	40	75	52100	0.556	0.541
	1200	2	40	75	52100	0.752	0.970
MLO 75-8	600	1	1	75	52100	0.157	0.152
			10	75	52100	0.259	0.246
			40	75	52100	0.597	0.561
MLO 75-9	600	1	1	75	52100	0.155	0.150
			10	75	52100	0.259	0.244
			40	75	52100	0.549	0.559
MLO 75-10	600	1	1	75	52100	0.165	0.150
			10	75	52100	0.254	0.236
			40	75	52100	0.526	0.541
MLO 75-11	600	1	1	75	52100	0.173	0.152
			10	75	52100	0.251	0.244
			40	75	52100	0.351	0.336
MLO 75-16	600	1	1	75	52100	0.180	0.163
			10	75	52100	0.251	0.229
			40	75	52100	0.587	0.564
MLO 75-17	600	1	1	75	52100	0.157	0.191
			10	75	52100	0.251	0.241
			40	75	52100	0.615	0.584
MLO 75-18	1200	1	40	75	52100	0.589	0.671
						0.566	0.645
MLO 72-25	600	2	10	75	52100	0.297	0.257
					M-10	0.249	0.224
			40	75	52100	0.462	0.493
					M-10	0.391	0.417
MLO 75-26	600	2	10	75	52100	0.241	0.236
					M-10	0.231	0.249
			40	75	52100	0.488	0.500
					M-10	0.356	0.376
MLO 75-27	600	2	10	75	52100	0.239	0.239
					M-10	0.246	0.236
			40	75	52100	0.495	0.493
					M-10	0.348	0.353
MLO 75-51	1200	1	4	75	52100	0.485	0.478
			10	75	52100	0.427	0.437
			20	75	52100	0.531	0.516
MLO 75-56	1200	1	4	204	52100	0.638	0.500
			10	204	52100	0.759	0.729
			20	204	52100	0.899	1.135
MLO 75-56A	1200	1	4	75	52100	0.470	0.485
			10	75	52100	0.544	0.594

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
			20	75	52100	0.533	0.536
MLO 75-111	600	1	1	75	52100	0.234	0.246
			10	75	52100	0.556	0.556
			40	75	52100	0.681	0.635
MLO 75-111A	600	2	10	75	52100	0.561	0.513
MLO 25-120	600		40	75	52100	0.757	0.759
MLO 75-121	1200	1	1	75	52100	0.234	0.236
			10	75	52100	0.345	0.363
			40	75	52100	1.491	1.161
MLO 76-12	600		40	75	52100	0.724	0.709
MLO 76-21	1200	1	1	75	52100	0.234	0.249
			10	75	52100	0.361	0.361
			40	75	52100	0.818	0.818
MLO 76-23	600	1	1	75	52100	0.175	0.180
			10	75	52100	0.246	0.305
			40	75	52100	0.523	0.528
MLO 76-24	600	1	1	75	52100	0.170	0.132
			10	75	52100	0.239	0.246
			40	75	52100	0.437	0.490
MLO 76-27	1200	1	1	75	52100	0.236	0.213
			10	75	52100	0.399	0.419
			40	75	52100	1.207	1.006
MLO 76-28	1200	1	1	75	52100	0.152	0.142
			10	75	52100	0.249	0.274
			40	75	52100	0.879	1.372
MLO 76-29 +1% MLO 69-48	1200	1	40	75	52100	1.011	0.963
MLO 76-29 +2% MLO 69-48	1200	1	40	75	52100	1.153	1.110
MLO 76-39	600	1	1	75	52100	0.170	0.150
			10	75	52100	0.259	0.246
			40	75	52100	0.594	0.584
	1200	1	1	75	52100	0.226	0.297
			10	75	52100	0.345	0.351
			40	75	52100	0.627	0.610
MLO 76-46	1200	1	1	75	52100	0.257	0.269
			10	75	52100	0.389	0.366
			40	75	52100	0.927	0.907
MLO 76-74	1200	1	40	75	52100	1.138	1.125
						2.342	1.311
MLO 76-74 +2% A	1200	2	40	75	52100	1.567	1.359
MLO 76-74/TCP	1200	1	40	75	52100	1.405	1.306
MLO 76-74 +5% A	1200	2	40	75	52100	1.412	1.196
MLO 76-92	1200	1	1	75	52100	0.234	0.203
			10	75	52100	0.358	0.333
			40	75	52100	0.597	0.610
MLO 76-98	1200	1	1	75	52100	0.168	0.183

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
			10	75	52100	0.244	0.236
			40	75	52100	0.472	0.597
MLO 76-107	1200	1	40	75	52100	1.473	1.440
MLO 76-108	60	1	40	75	52100	0.648	0.592
MLO 76-122	1200	1	40	75	52100	0.907	1.255
MLO 76-126	600	1	1	75	52100	0.114	0.201
			10	75	52100	0.226	0.226
			40	75	52100	0.323	0.368
MLO 76-127	600	1	1	75	52100	0.160	0.175
			10	75	52100	0.267	0.231
			40	75	52100	0.531	0.521
MLO 76-136	600	0.17	75	200	440-C		0.520
		0.22	75	200	440-C	3203	
		1.5	10	75	440-C	0.300	0.269
					52100	0.645	0.640
		2	10	75	52100	0.714	0.704
			40	75	52100	0.853	0.902
				200	52100	0.986	1.034
	1200	0.08	75	200	440-C	4.983	9.938
		0.33	40	200	440-C	5.819	
		2	10	75	440-C	0.307	0.323
				200	400-C	1.676	1.661
						1.422	1.400
					52100	0.432	0.391
			40	75	440-C	3.891	4.262
					52100	0.988	0.927
				200	440-C	5.070	3.447
							3.112
			75	75	52100	5.237	5.055
				200	52100	3.005	2.781
MLO 76-137	600	1.5	10	75	440-C	0.635	0.643
		2			52100	0.505	0.523
		2	10	200	52100	0.655	0.815
			40	200	52100	0.899	0.945
			75	200	52100	0.927	1.161
	1200	1	40	75	52100	0.561	0.566
						0.559	0.564
					52100/TCP	0.559	0.531
						0.549	0.531
		2	10	200	440-C	1.384	1.402
					52100	0.706	1.214
						0.592	1.118
						0.673	0.688
			40	75	440-C	2.385	2.664
				200	440-C	2.380	2.421
						2.454	2.273
					52100	1.123	1.374

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
						0.953	1.427
			75	75	52100	1.046	1.542
				200	440-C	2.824	3.035
						2.710	3.175
					52100	1.443	1.549
						1.339	1.356
MLO 76-137	600	2	10	200	52100	0.665	0.721
Degassed			40	200	52100	0.947	0.980
			75	200	52100	1.082	1.031
MLO 77-041	1200	2	40	75	52100	0.742	0.838
MLO 77-47	1200	1	40	75	52100	0.963	0.919
MLO 77-49	1200	1	10	75	52100	0.665	0.523
						0.671	0.536
					52100/TCP	0.594	0.544
						0.693	0.536
			40	75	52100	0.859	0.892
						0.818	0.757
						0.940	0.795
						0.861	0.810
					52100/TCP	0.886	0.879
						0.889	0.925
MLO 77-49	600	2	10	200	52100	0.467	0.500
Degassed			40	200	52100	1.049	1.255
			75	200	52100	1.059	1.748
MLO 77-51	600	1	1	75	52100	0.152	0.175
			10	75	52100	0.254	0.226
			40	75	52100	0.511	0.368
MLO 77-66	600	1	1	75	52100	0.160	0.163
			10	75	52100	0.262	0.257
			40	75	52100	0.577	0.579
MLO 77-70	1200	1	1	75	52100	0.160	0.183
			10	75	52100	0.284	0.300
			40	75	52100	0.841	0.897
MLO 77-99	1200	1	40	75	52100	0.904	0.968
MLO 77-121	1200	1	1	75	52100	0.292	0.351
			10	75	52100	0.584	0.610
			40	75	52100	0.841	0.864
MLO 77-122	1200	1	1	75	52100	0.147	0.157
			10	75	52100	0.239	0.290
			40	75	52100	0.818	0.742
MLO 77-123	1200	1	1	75	52100	0.284	0.284
			10	75	52100	0.340	0.328
			40	75	52100	0.810	0.734
MLO 77-124	1200	1	1	75	52100	0.279	0.318
			10	75	52100	0.480	0.584
			40	75	52100	0.714	0.805
MLO 77-125	1200	1	1	75	52100	0.147	0.193
			10	75	52100	0.234	0.254

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
			40	75	52100	0.617	0.627
MLO 77-126	600	1	10	75	440-C	0.292	0.279
					52100	0.559	0.[illegible]41
		1.5	10	75	440-C	0.389	0.290
					52100	0.610	0.655
		2	10	75	440-C	0.277	0.295
					52100	0.688	0.686
				200	440-C	0.953	0.742
					52100	0.427	0.376
			40	75	440-C	3.073	3.040
					52100	0.925	0.932
			75	75	52100	0.996	0.942
	1200	1	40	75	52100	0.772	0.635
						0.630	0.699
					52100/TCP	0.686	0.813
						0.767	0.869
MLO 77-127	1200	1	40	75	52100	0.721	0.775
						0.749	0.752
					52100/TCP	0.762	0.775
						0.772	0.795
MLO 77-144	1200	1	40	75	52100	0.714	0.757
						0.711	0.714
						0.693	0.765
						0.709	0.757
					52100/TCP	0.640	0.714
						0.711	0.841
MLO 77-145	1200	1	40	75	52100	0.627	0.439
						0.612	0.478
						0.607	0.506
						0.505	0.419
					52100/TCP	0.579	0.470
						0.622	0.472
MLO 77-146	1200	1	40	75	52100	0.445	0.417
						0.455	0.411
MLO 77-147	1200	1	40	75	52100	0.787	0.650
						0.777	0.856
						0.655	0.815
						0.633	0.665
MLO 78-29	1200	1	40	75	52100	0.480	0.470
MLO 78-30	1200	1	40	75	52100	0.787	0.777
MLO 78-31	1200	1	40	75	52100	1.135	1.163
MLO 78-32	1200	1	40	75	52100	2.926	2.456
MLO 78-33	1200	1	40	75	52100	1.326	1.252
MLO 78-34	1200	1	40	75	52100	0.704	0.757
MLO 78-35	1200	1	40	75	52100	0.831	0.848
MLO 78-36	1200	1	40	75	52100	0.884	0.907
MLO 78-37	1200	1	40	75	52100	0.968	0.919
MLO 78-38	1200	1	40	75	52100	0.874	1.003
MLO 78-39	1200	1	40	75	52100	0.620	0.650
MLO 78-40	1200	1	40	75	52100	1.605	0.747

Table 2 (continued)

Lubricant	Speed (rpm)	Time (hr)	Load (kg)	Temp. (°C)	Ball Material	Wear Scar (mm) Rig 2	Rig 3
MLO 78-41	1200	1	40	75	52100	1.424	1.453
MLO 78-42	1200	1	40	75	52100	1.374	1.415
MLO 78-46	1200	1	40	75	52100	0.808	0.782

Note: AMB denotes ambient laboratory air temperature, usually 20-25°C. These tests are run for 4 hours to allow the operating temperature to stabilize.

SECTION IV

GREASE TESTING ON THE POPE SPINDLES

After the preliminary screening tests, such as the Four-Ball wear test, selected lubricants are tested in actual machine elements. One of these tests is the Pope spindle test, in which the candidate lubricant (generally a grease) is packed in a size 204 bearing. The bearing is rotated at 10,000 rpm on an operating cycle of 20 hr rotation and 4 hr nonrotation. The test temperature is controlled at the required value during rotation; no heat is applied during the nonrotating portion of the cycle. Lubricant performance is measured by the operating life of the bearing under these conditions. Lubricant failure can be by bearing over-temperature (frictional heating of the bearing) or by excessive drive motor current (excessive frictional torque) either during operation or at start-up after the nonrotating portion of the cycle.

This section of the report contains a description of the Pope spindle, gives results of the testing effort, and presents conclusions that can be drawn as a result of this work.

A. Equipment Description

The Pope spindle has two size 204 ball bearings mounted on a rotating shaft; one bearing is the test bearing while the second is a support bearing. The experimental bearing is packed with 5 cc of the experimental grease. The grease is installed in and on the bearing by using a hypodermic syringe, assuring an even distribution of lubricant over all the bearing surfaces.

The rotating shaft, mounted horizontally, is belt-driven at 10,000 rpm by a step-up pulley from a 3,500-rpm drive motor. Axial loading of the test bearing is accomplished by using wavy washers (axial force springs); radial load is controlled by using part of the weight of the motor acting on the spindle through the driving belt. The loads used for this work were 22 N axial and 22 N radial. A schematic of the Pope spindle is shown in Figure 5.

On the left end of the rotating spindle are mounted the left end flinger and the test bearing. The bearing is seated against a shoulder on the spindle. On the right end of the spindle are mounted the support bearing, seated against the shoulder opposite the test bearing, and the driven pulley. End caps, attached to the spindle housing, position the bearing outer races, the outer seals, and the wavy washer.

The shoulder area of the spindle, separating the test and support bearings, also serves as the mating surface for the seals separating the test

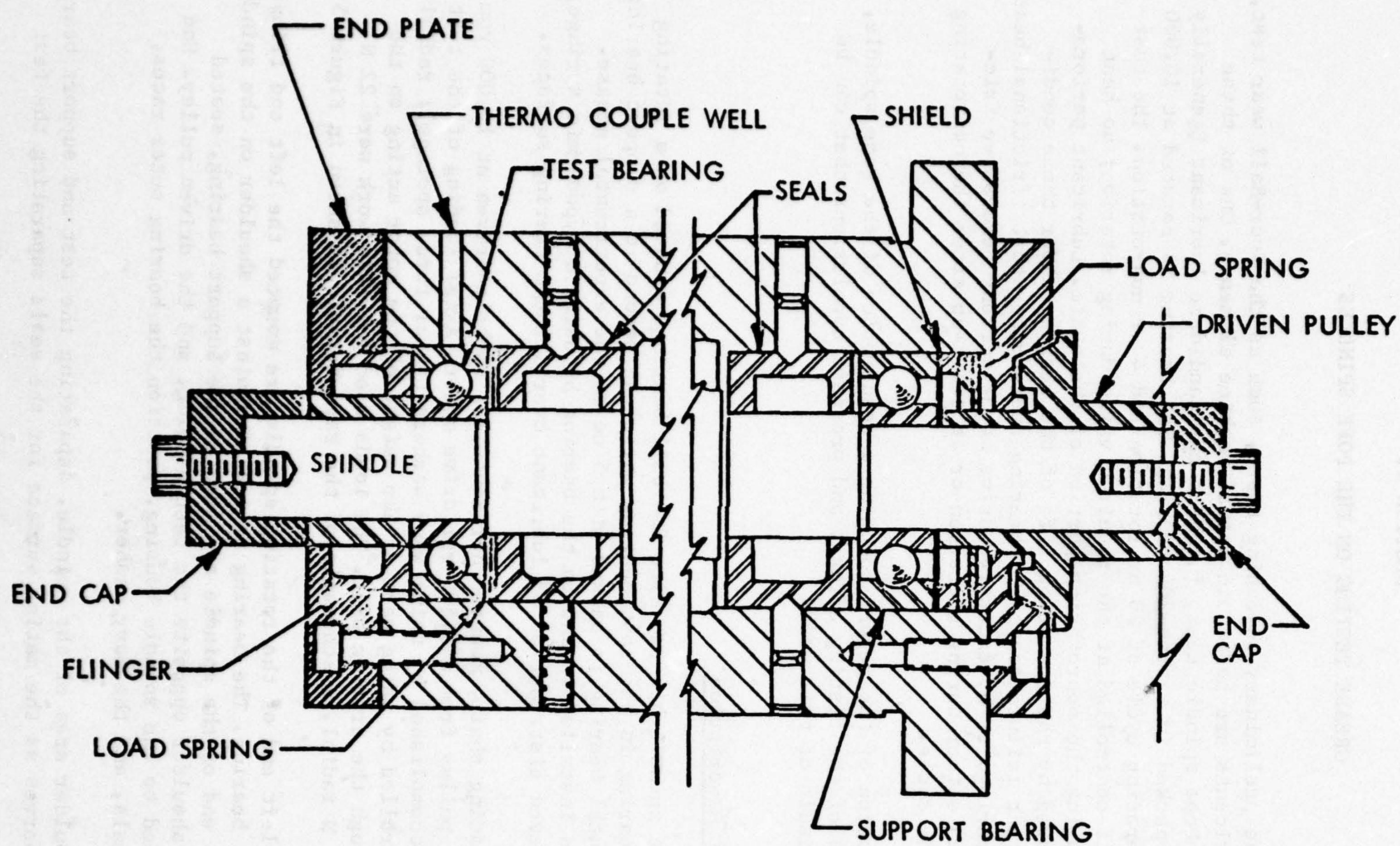

Figure 5 - Pope Spindle Schematic

area from the support area. The support bearing is lubricated with conventional greases; adequate seals and reservoir volumes, coupled with the high rotational speeds, keep the support bearing lubricant from contacting the test bearing.

The spindle assembly fits into a mounting bracket on the machine base. The base holds the pivot-mounted and spring-supported motor plate and motor. There are provisions for adjusting the amount of spring force necessary to provide the required belt tension for the specified radial load on the spindle assembly. Motor speed controls (high, low, off), heater controls, and temperature-monitoring provisions are also included on the machine base. The unit is limited to operating in a normal laboratory air atmosphere.

B. Results

The results of the 27 experiments on 20 different greases are presented in Table 3.

None of the greases provided lubrication to the test bearing for the 2,000-hr requirement on the Pope spindle. Only two greases, MCG 76767042 and MCG 77031432, could even be considered as coming close to the 2,000-hr goal. The two experiments using these greases were stopped for grease inspections at 1,530.1 and 1,061.2 hr, respectively.

C. Conclusions

Based upon the samples tested, work seems to be required to provide greases capable of meeting the requirements of this 2,000-hr test.

TABLE 3

RESULTS OF GREASE TESTING ON THE POPE SPINDLES

Lubricant	Temperature (°C)	Operating Time (hr)	Remarks
MCG 70-27	177	70.5	Grease Failure
MCG 70-28	177	117.6	Grease Failure
MCG 75186021	177	338.8	Grease Failure
		354.2	Grease Failure
MCG 75245026	177	257.1	Grease Failure
		1,183.4	Grease Failure
MCG 76189034	232	282.8	Grease Failure
		210.0	Grease Failure
MCG 76267042	232	1,107.9	Grease Failure
		1,530.1	Stopped for Inspection
MCG 77011204	232	45.3	Grease Failure
	260	48.3	Grease Failure
MCG 77020112	232	23.9	Grease Failure
	260	23.4	Grease Failure
MCG 77031432	232	1,061.2	Stopped for Inspection
MCG 77081663	204	121.4	Grease Failure
MCG 77081664	204	323.6	Grease Failure
MCG 77092970	149	1,562.[illegible]	Grease Failure
MCG 77092971	149	1,614.5	Grease Failure
MCG 78010401	149	466.3	Stopped for Inspection
MCG 78010402	149	188.5	Grease Failure
MCG 78010403	149		Still Running
MCG 78010404	149		Still Running
MCG 78010405	149		Still Running
MCG 78010406	149	371.8	Grease Failure
MCG 78012407	177	347.7	Grease Failure
		625.5	Grease Failure

Test Conditions: Radial load - 22N
Axial load - 22N
Bearing size - 204
Operating speed - 10,000 rpm
Operating cycle - 20 hr on, 4 hr off
Operating environment - laboratory air

SECTION V

GREASE TESTING ON THE NAVY SPINDLES

The Navy spindle is similar to the Pope spindle in that both are used to conduct screening tests using actual machine elements. The Navy spindle uses a size 204 test bearing, operating at 10,000 psi, as does the Pope spindle. The Navy spindle operating cycle is 20 hr of rotation, followed by 4 hr of nonrotation, like the operating cycle of the Pope spindle. The Navy spindle employs a 22-N thrust load as does the Pope spindle. The Navy spindle uses a 22-N radial load; the Pope spindle is normally limited to a 22-N radial load. Operating lives for the Navy spindle are reported in cycles of operation (one cycle being 20 hr of rotation followed by 4 hr of nonrotation), while the Pope spindle operating lives are reported as operating time in hours. The criterion of passing the test is still 2,000 hr of operation, or 100 cycles.

A. Equipment Description

The Navy spindle uses one size 204 test bearing as does the Pope spindle. In the Navy spindle, the shaft is driven at 10,000 rpm through a belt drive from a 3,450-rpm motor. The spindle is supported by a size 206 bearing on the motor end of the shaft and a size 305 bearing on the test end of the shaft. The size 204 test bearing is mounted on the shaft out-board of the size 305 support bearing. A schematic of the Navy spindle test zone is presented in Figure 6.

The test bearing inner race is mounted on the rotating shaft, and the outer race is installed in a holder, to which are attached the radial load deadweights and the two outer race thermocouples. One thermocouple is for the temperature controller and one is for the temperature readout. No heat was applied to the test bearing in this work. The thrust load is applied to the outer race from the basic spindle framework through the thrust load spring.

As the frictional torque in the test bearing increases, the outer race, holder, and radial load hanger tend to rotate. If this fractional torque exceeds 0.141 N•m, the radial load hanger stops the test. Such shutdown terminates the experiment.

B. Results

Only four Navy spindle tests were conducted during the contract period, and all were conducted in laboratory ambient air conditions. The results of these tests are presented in Table 4.

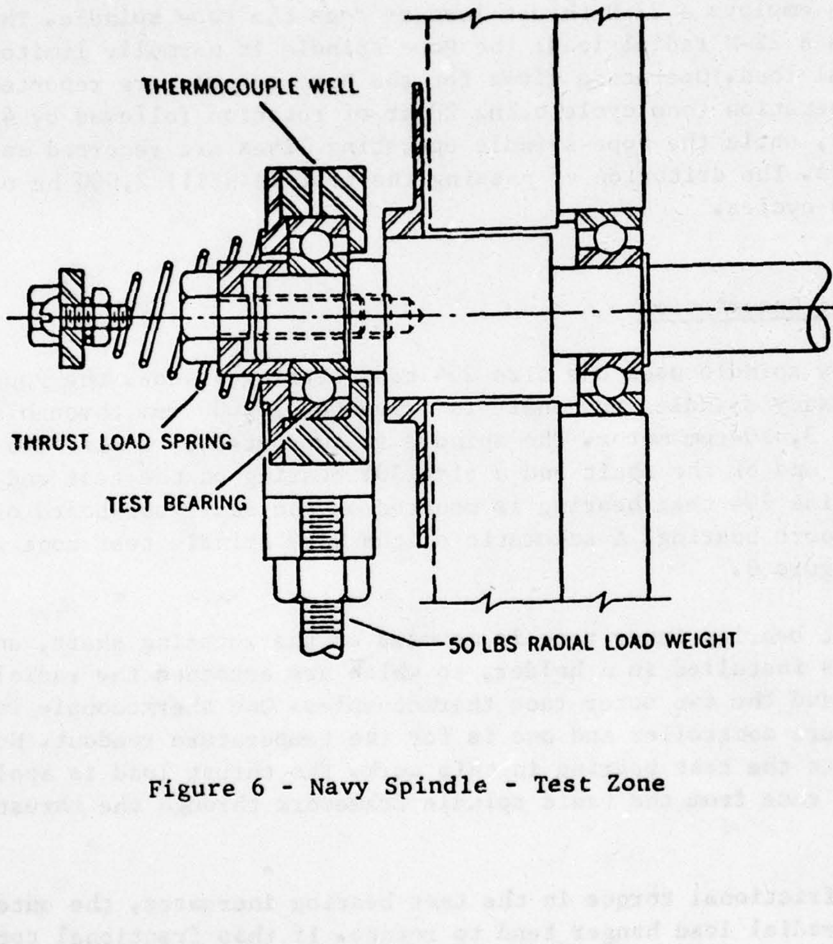

Figure 6 - Navy Spindle - Test Zone

TABLE 4

RESULTS OF GREASE TESTING ON THE NAVY SPINDLES

Test Conditions: Radial load: 222N
Axial load: 22N
Bearing size: 204
Operating speed: 10,000 rpm
Operating cycle (1): One cycle is 20 hr rotation; 4 hr nonrotation
Operating environment: Laboratory air

Lubricant	Temperature (°F)	Operating Life (Cycles) (1)	Remarks
MCG 731995	204	32	Stopped for inspection
		14	Stopped due to failure of support bearing
		18	Grease failure
		93	Stopped due to failure of support bearing

The attempts made to test this grease revealed that during either recent equipment relocations or equipment modifications, misalignment between the support bearings had become a problem.

The value of these tests was re-examined in light of the equipment problems. The decision was made to discontinue this area of activity for the present time.

C. Conclusions

The expense and complexity of realigning the housings of the support bearings were determined to be more than the value of the test results obtained, and this tester-type has been retired.

SECTION VI

GREASE TESTING ON THE SIKORSKY RIGS

The Sikorsky Aircraft Friction Oxidation Tester, Model SKP-172-1, hereinafter called the Sikorsky rig, was established as a tester for qualifying greases to USAF Military Specification MIL-G-25537A. The Sikorsky rig was to subject candidate greases to conditions similar to helicopter applications in which fretting corrosion could be a hazard.

This part of the report contains a description of the rig, the results of the testing that has been done during the past 3 years, and a section on conclusions and comments that can be made about both the results and the Sikorsky rig test method.

A. Equipment Description

Ths Sikorsky rig subjects two tapered roller bearings to both axial and radial loads under low-amplitude oscillating motion (fretting conditions). The inner cones of the bearings are oscillated at 410 cycles/min through an arc of $\pm$ 3 degrees (6 degrees total motion). Lubricant performance is measured by the increase in frictional torque of the bearings, and a lubricant is considered to have passed the test if the measured frictional torque does not exceed 56.6 N•m (500 in.-lb) after 250 hr of operation.

The test bearing assembly, shown schematically in Figure 7, consists of two Timken test bearings (2,631 cups and 2,687 cones) and two RBC ESJ 7295 needle bearings (load bearings) on the SKP-746 shaft. The 2,631 cups of the test bearing are mounted in the two upright portions of the basic assembly. The thrust plate separates the two uprights and maintains the parallel alignment of the cups when the axial and radial loads are applied. The outer races of the needle bearings are mounted in the loading yoke assembly that transmits the 44.5-kN (10,000-lb) total radial load onto the test bearings.

The axial load on the tapered roller test bearings comes from the torque requirement of 68 N•m (50 ft-lb) on the 5/8-18 nut on the crank arm end of the SKP-746 shaft. The axial load is, therefore, a function of the thread-to-thread (nut-to-shaft) friction and the nut face-to-cone surface friction. If there were no friction in the threads and nut mating surfaces, the axial load would be 302 kN (67,858 lb). For a friction coefficient of 0.1, the axial load would be 272 kN (61,072 lb); for a friction coefficient of 0.5, the load would be 151 kN (33,929 lb); and for a friction coefficient of 0.9, the axial load would be 30 kN (6,786 lb). There is no thread or mating surface friction determination made as part of the specification

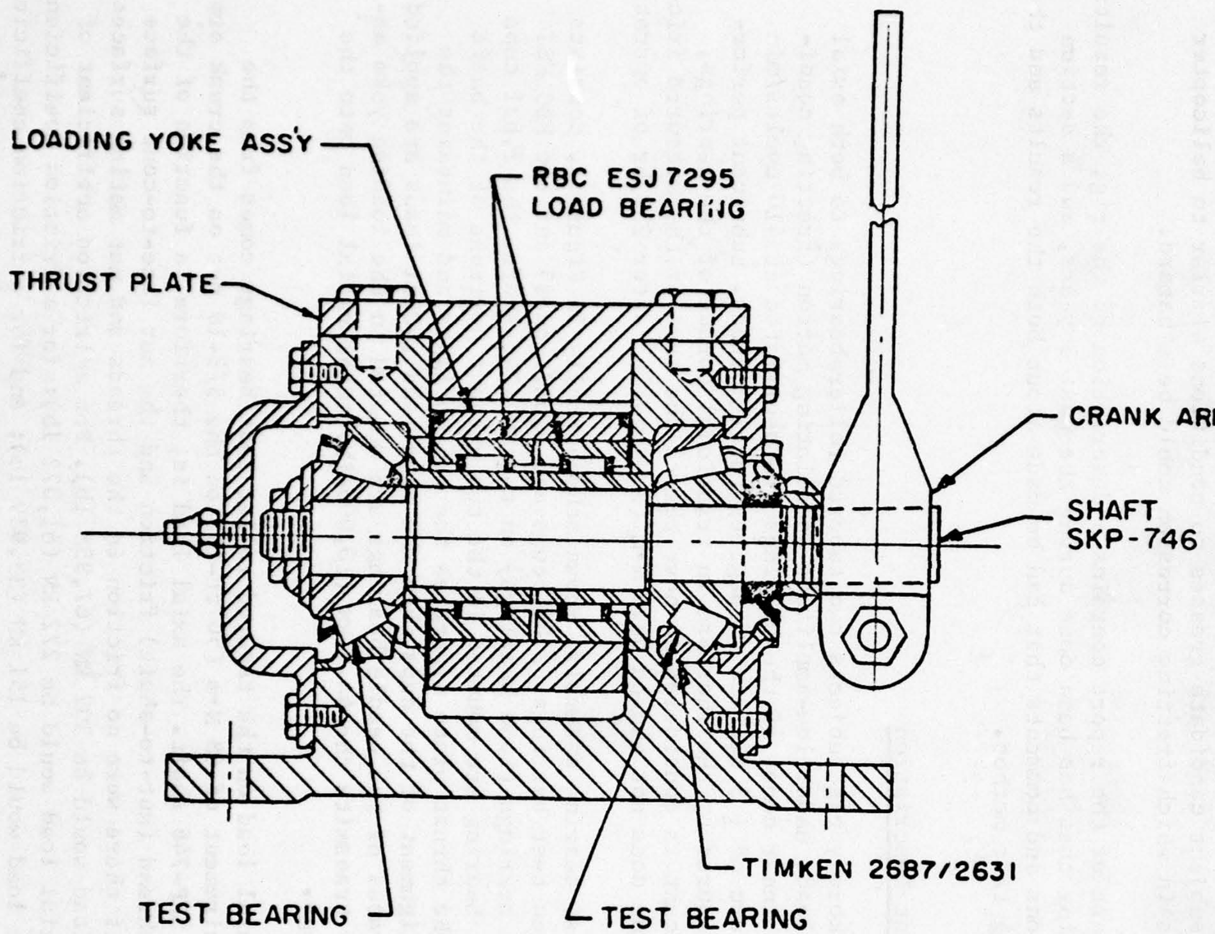

Figure 7 - Sikorsky Rig Test Bearing Assembly

test procedure; therefore, the axial load on the tapered roller bearings during the test is not known.

The test bearing assembly is one part of the three basic systems in the Sikorsky rig. The other parts of the system, shown in Figure 8, are the hydraulic system and the driving mechanism. Radial loading of the test bearings is done by the hydraulic system. Oil is pumped from the reservoir (A), with the hand pump (B), to the loading cylinder (E). When the desired pressure has been obtained, as determined by a pressure gauge, shown at (D), the accumulator (C) maintains the hydraulic pressure throughout the test. The valve (K) is closed after the correct pressure [10 MN/m^2 (1,500 psi)] has been established, thereby eliminating leakage back through the hand pump. The radial force on each of the test bearings is 22 kN (5,000 lb). The loading cylinder (E) pushes on the loading yoke, which is part of the test bearing assembly (J). A variable speed electric motor (F) provides the drive for the test through a speed reducer and a Krouse adjustable throw crank (G). The crank is connected to the crank arm assembly by the lever arm (H) which includes a proof ring for the measurement of the frictional drag of the test bearing assembly. After the test has been completed, the valve (L) is opened and the hydraulic fluid is returned to the reservoir.

In the operating procedure (Ref. 7) and in the specification (Ref. 8), the same criteria for establishing the axial load are described. The procedure is to tighten the 5/8-18 nut to 61 to 68 N•m (45 to 50 ft-lb) of torque and check to see if the frictional drag is between 133 and 156 N (30 and 35 lb) at the crank arm assembly. If the drag is low, the torque is to be increased to 68 N•m (50 ft-lb); if the drag is high, the torque is to be decreased to 61 N•m (45 ft-lb). If the drag is still not within limits, the test bearing assembly is to be disassembled and reassembled to try again. One interpretation that can be placed upon this type of an operating instruction is that the better the grease (in friction reduction), the greater the load exerted on the test bearings, and conversely.

The criterion of lubricant failure is an increase in frictional drag to 267 N (60 lb) at the crank arm assembly, which is 56.5 N•m (500 in.-lb) of torque in the test bearing assembly.

B. Results

The results of the Sikorsky rig testing are presented in Table 5. As can be seen from the table, 37 materials were subjected to 51 tests. Two tests, one on each of two machines, are required for qualification. If both machines operate for 250 hr or more, the material is considered to have passed the test. A rerun can be made if the test fails before 250 hr of operation. Operating times shown in Table 5 that are over 250 hr do not mean that the grease failed at the time shown in the table; tests were often terminated before grease failure.

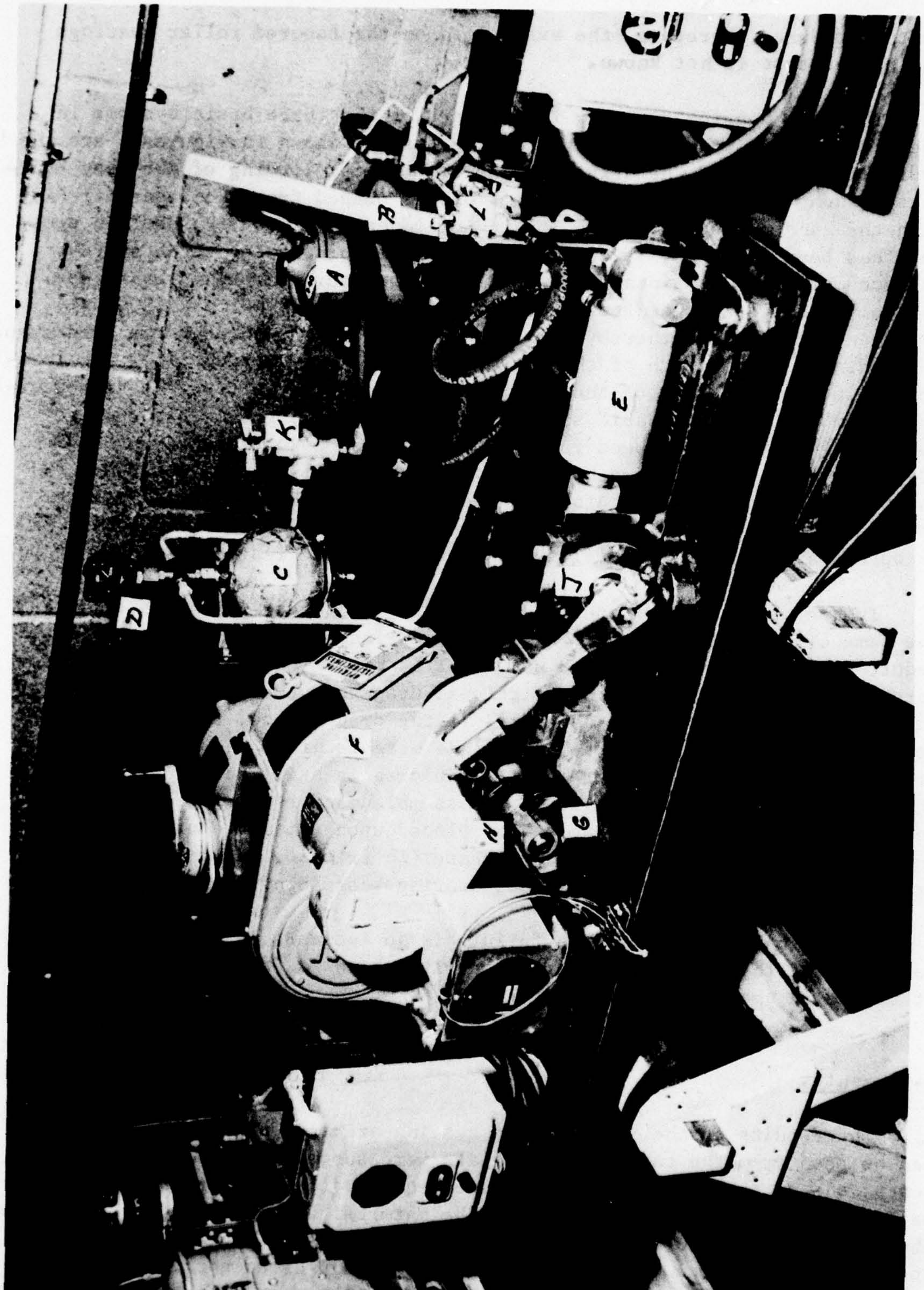

Figure 8 - Sikorsky Rig

TABLE 5

RESULTS OF GREASE TESTING IN THE SIKORSKY RIGS

Lubricant	Operating Time, hr[a/] Machine No. 4	Machine No. 5
MCG 75058001	282.9	282.7
MCG 75252027	299.3	139.0
MCG 75276028	1.0	1.0
MCG 75332039	162.0	55.0
MCG 76015011	2.0	2.0
MCG 76015012	2.0	2.1
	3.0	3.0
MCG 76050015	16.0	16.0
MCG 76103018	305.0	
MCG 76113020	258.9	
MCG 76113021	229.0	
MCG 76138030		202.0
MCG 76188033	130.0	
MCG 76313046	19.1	
MCG 76313047		19.0
MCG 76313048	19.2	
MCG 76313049		19.2
MCG 76313050	151.0	
MCG 76324059		248.0
MCG 77010001		263.5
MCG 77020313		18.8
MCG 77020314		69.1
MCG 77020315	155.6	
MCG 77020316	176.8	
MCG 77020317		142.5
MCG 77021423	113.2	
MCG 77021424	259.0	
MCG 77030430		92.1
MCG 77052436	262.2	262.2
MCG 77061643	20.0	20.0
MCG 77072646	246.0	
MCG 77072647		234.0
MCG 77080150	246.0	234.0
MCG 77092967	5.4	
MCG 77092968		5.3
MCG 77092969	5.8	5.8
MCG 77120585	261.9	260.7
MCG 78010406		2.2

a/ An operating time of > 250 hr is required to pass the test.

Test Conditions: Radial load = 22 kN, each bearing
Cyclic speed = 410 cycles/min
Axial load = Not known (between 0 and 68,000 lb)
Arc of motion = ± 3 degrees

Three greases passed the Sikorsky test--MCG 75058001, MCG 77052436, and MCG 77120585. Four others, in which only single tests were performed, exceed the 250-hr requirement and may have qualified if the duplicate tests would have been run on the second rig. These greases were: MCG 76103018, MCG 76113020, MCG 77010001, and MCG 77021424.

Tests are not always run in duplicate because considerable operator time is required for set-up and tear-down; hence, more materials can be tested if only one machine per experimental grease is used. Those greases which fail early in the 250-hr test do not warrant testing on the second machine unless justified from other considerations.

C. Conclusions

1. One conclusion that can be drawn from the operational performance of the greases is that three of the 37 candidate greases passed the Sikorsky fretting corrosion (or oxidation) test.

2. The primary cause of the increase in frictional drag of the test bearings is due to the roller-cone wear. This wear allows the rollers to ride back on the cone, where the rollers contact the shoulder at the end of the conical section. The end of the roller rubs the flat face of the shoulder, increasing the frictional drag greatly over the relatively simple rolling contact of the normal bearing.

Other conclusions can be made relative to the test method and apparatus.

3. The first of these "procedure" conclusions is that there seems to be no way of knowing the axial load actually experienced by the test bearings.

4. Following the established operating procedure of adjusting the torque on the axial load nut until the drag is within limits penalizes the greases that permit the bearing to operate with low friction. The procedure requires that the torque (hence, axial load) be increased if the drag is less than 133 N (30 lb) at the crank arm assembly.

5. The nature of the axial load nonadjustment during operation benefits the grease that allows greater wear in the test bearing. When fretting of the tapered roller bearing occurs, the shaft axial load relaxes as the shaft elongation (hence, tension stress) is reduced. The complete converse is not true, but the less wear produced in the test bearing, the greater the magnitude and duration of the axial load.

6. This test procedure and apparatus are suited for qualification testing of candidate greases to a specific application, but their use in research on the operational capacity of greases is limited.

SECTION VII

STUDIES ON LUBRICANT COMPACTS IN BALL BEARINGS

This part of the report presents operational data on three different lubricant compact materials in ball bearings and extends the data reported in References 1 and 2. These compact materials, formed by powder metallurgy techniques, include metal powders for structural strength and lubricating powders. Lubricant compacts were developed for operation in ball bearings that must operate without oil or grease lubricants. Five size 204 ball bearings, modified to include the lubricant compact material inserts in the ball separators, have been operated throughout the duration of this contract. Of these five bearings, the lubricant compacts have provided satisfactory operation from 31,075 hr to more than 94,024 hr. The longest operating time for a lubricant compact bearing is 94,024 hr and, at this writing, the bearing is still operating satisfactorily.

The bearings were installed in individual operating chambers and subjected to a vacuum environment and a 31-N axial load. Rotation of the bearings was provided by standard 1,800-rpm electric motors, which were found to rotate at 1,790 rpm under load. A magnetic coupling, established through the bottom of each of the test chambers between the driving motor and driven bearing, was used to rotate the test bearing. Coast-down time measurements were made to determine the frictional torque of the test bearing. Periodic bearing-weight determinations were made to determine the loss of weight of compact material.

The following paragraphs of the report include a description of the equipment and experimental bearings used in this work, results of the experiments, wear rates of the compacted materials, predicted wear-lives for each of the bearings, and some of the conclusions that can be drawn from the work.

A. Equipment Description

Five bearing rigs were used for this work. Each rig is capable of subjecting one size 204 ball bearing to a light load in a vacuum environment. Each of the rigs is a separate operating station, consisting of test chamber, bearing holder, drive motor, magnetic coupling, and vacuum pump. The loading of the bearing, a nominal 31-N axial load, is generated by the weight of the rotating portion and the attraction between two magnets. One magnet, inside the chamber, is attached to the rotating bearing inner race through a shaft. The other magnet is outside the chamber and attached to the driving motor. A schematic of one of the rigs is shown in Figure 9.

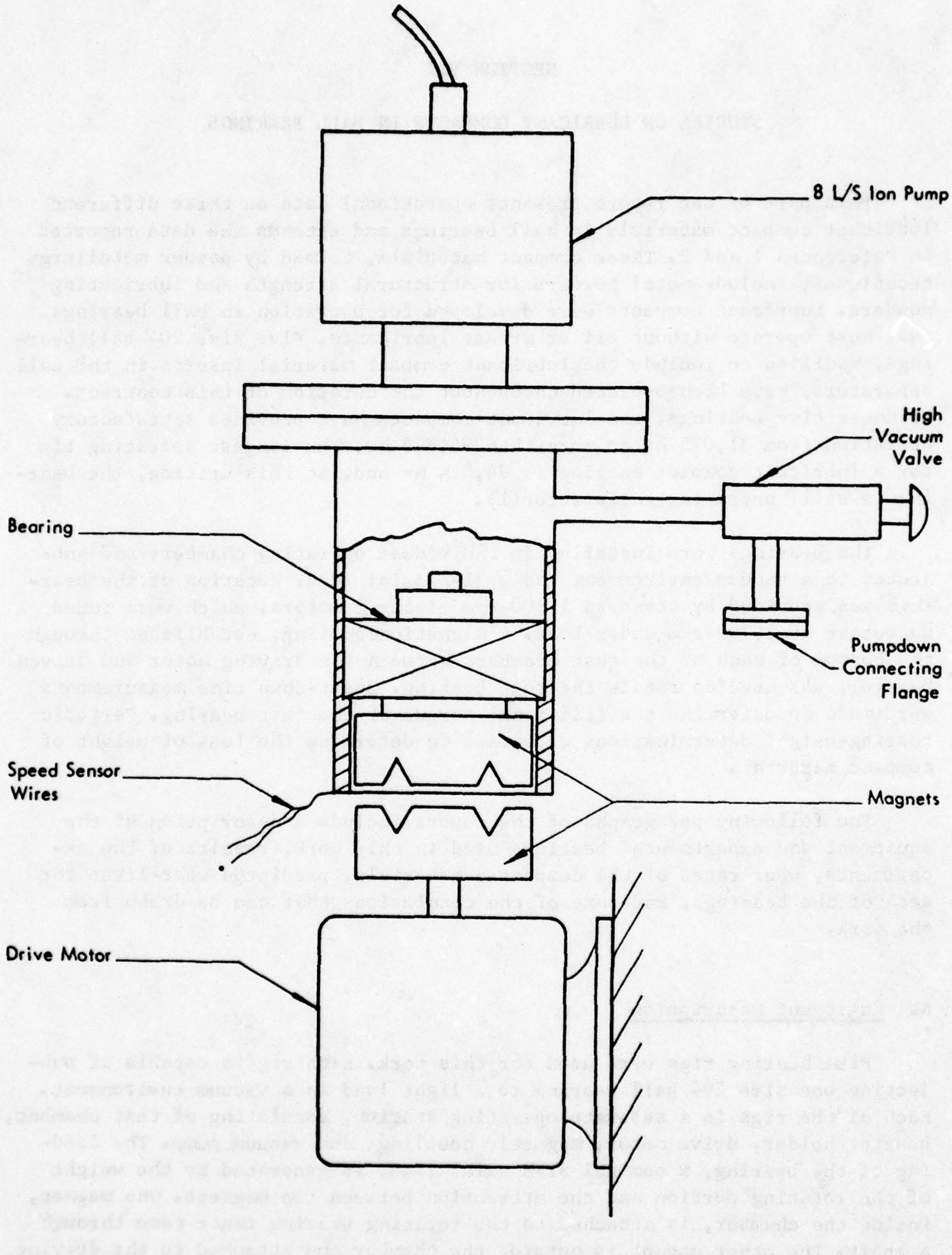

Figure 9 - Long-Term Bearing Rig Test Schematic

The rotating portion of the load consists of the rotating shaft, spacer, washer, drive magnet, another washer, and the retaining screw, all of which are attached to the bearing inner race. The bearing outer race rests on a shoulder of the inner cylinder, which in turn is held in place by the long spacer, shield, and screw ring. The inner cylinder fits into the test chamber, is held in place by the snap ring, and is retained from rotating by a pin inside the chamber that fits into a slot in the inner cylinder.

The test chamber is attached to the ion pump; the copper gasket is used to form the vacuum seal. The assembly fits into a holder which positions the test chamber over the driving motor and magnet.

The driving motor and magnet can be raised and lowered, allowing the magnetic coupling between the driving and driven magnets to be established through the bottom of the test chamber. A rheostat is used in the driving motor circuit so that motor speed can be increased gradually. After full speed of the motor has been reached, the rheostat is removed from the circuit and full-line voltage powers the motor. The full speed of the motor, under load, has been measured as 1,790 rpm.

Bearing frictional torque is determined from the coast-down time of the bearing in the chamber. Coast-down time is measured by loosening a holding screw and quickly lowering the motor and driving magnet approximately 50 mm. A voltage, induced in a coil wrapped around the lower end of the test chamber by the rotation of the driven magnet, decays to zero as the rotating portion comes to rest. The time between motor lowering and voltage decay to zero is the coast-down time. Frictional torque has been calculated and plotted as a function of coast-down time (Ref. 9). If the bearing frictional torque exceeds 0.31 N•m, the magnetic coupling torque is exceeded and the bearing cannot be driven.

The magnetic attraction between the driven and driving magnets contributes to the load on the test bearing. This magnetic force is measured before bearing rotation begins and after the experiment has been completed, if the bearing has not been destroyed. The eight-pole magnets are indexed at the four possible alignment positions. At each position, the driving magnet is loaded with deadweight until the attractive force is overcome. Then, the breakaway force, the deadweight, is measured. The force is determined twice for each location. The force recorded for the test bearing load is the average of these eight values added to the weight of the rotating portion (which is 5 N).

The experimental approach used for this work was to rotate the individual ball bearings in vacuum until the bearing frictional torque exceeded 0.31 N•m. Bearing friction was monitored daily and wear was determined periodically.

Prior to each introduction into the test chamber, each of the bearings was cleaned of foreign material by being "washed" with a gentle stream of dried nitrogen. No solvents were used in the cleaning process.

After being blown clean of wear debris and other contaminants, each bearing was weighed and then placed in one of the test chambers. The test chamber was then evacuated by using a sorption pump; the test pressure of nominally 1.33 μPa was sustained by an 8-liter/sec vac-ion pump. The bearing was loaded to a nominal 31-N axial load, and rotation was initiated. Operation of each bearing experiment was interrupted periodically to check frictional torque and/or bearing weight loss. The frequency of interruption depended upon factors such as expected bearing operating life, previous period weight loss, and coast-down time.

B. Test Bearing Description

The test specimens used for this work were all size 204 ball bearings, equipped with compacted material inserts in the ball separators. Two basic bearings were used. The earlier versions of the compact lubricant bearings were modified Fafnir MM204WI bearings; the later versions were New Departure SS30204DT.

The Fafnir MM204WI bearings, before modification, were ABEC-7 precision bearings (MM designation), size 204, light series, made of 52100 bearing steel. These bearings were of the maximum capacity type, with the outer race counterbored on one side for inclusion of the larger number (10) of larger diameter (8.731-mm or 11/32-in.) balls, as compared to the 200K series using eight balls of 7.938-mm (5/16-in.) diameter. The Fafnir bearings were used on two of the tests, one each with compacts AFSL-14 and AFSL-15.

The bearings used on the other three experiments were designed by New Departure for very heavy, unidirectional thrust loads. The contact angle for these bearings was 35 degrees, differing from the light thrust bearing series (15 degrees) and medium thrust bearing series (25 degrees) (Ref. 10). The New Departure bearing designation was SS30204DT. SS means that the balls, inner race, and outer race were made of 440C stainless steel. The 30204 denotes the 30,000 series bearings of high angular contact, in the size 204--bearing bore 20 mm, outer diamter 47 mm, and width 14 mm. The New Departure bearing had 10 balls of 8.731-mm (11/32-in.) diameter, as did the unmodified Fafnir MM204WI. The DT designation stands for "duplex tandem" and means that the bearing could be mounted in any of several configurations and, if used in tandem (with another bearing of the same designation), would have the designed value of preload established after assembly into the mounting fixture.

Development of the two separator designs used for these bearings has been previously described (Refs. 11 and 12). Both designs used only seven of the 8.731-mm (11/32-in.) diameter balls, equally spaced, in the lubricant compact retainers, to permit a greater possible wear volume for the lubricant inserts.

The earlier retainer used a cylindrical compact insert with a constant-diameter hole for the ball. The compact inserts were slightly over-size, so that the material could be preloaded in compression by interference fits between the inserts and the holes in the separator. Preliminary work (Ref. 13) had shown that the compacts had insufficient tensile strength to function as nonreinforced separators.

The later design of the bearing separator consisted of a cylinder 13.970 mm wide, with a nominal 30.734-mm inside diameter and 39.993-mm outside diameter, basically the same size as the earlier design. Both the holes for the inserts and the inserts themselves were made with a 9-degree taper, providing the compression preload without the cylindrical interference fit. The inserted material protruded slightly into the inner portion of the separator, leaving some compacted material (after finishing the separator inner bore) as the lubricant between the separator and bearing inner land, thus providing inner land control of the separator. The insert inner bores were 8.768 mm.

Two materials were used for the basic construction of the separators--2024 aluminum and 6Al4V titanium alloys. The aluminum alloy separators were designed to be used at room temperature only, while the more expensive titanium alloy separators were to be used primarily at elevated temperatures. Both types were used in this work to see if there would be any noticeable effect on performance due to separator material.

Three compacted materials were used as inserts for the test bearings. The materials and their compositions are presented in Table 6.

The development of these hot-pressed lubricant compacts has been described in previous reports and papers (Refs. 2, 12-16). A hot-pressed compact has been defined as a material formed by a powder metallurgy process to combine metal and lubricant powders into structural bodies. Briefly, the manufacture of these compacts started with the mixing of the proper weight ratios of the powdered materials, using either a ball mill or a solvent solution and blender for mixing. When the solvent solution was used, the solvent evaporated after mixing in the blender. The mixed powders were then placed in a carbon die and heated in a controlled inert atmosphere. After the proper temperature had been established, the die plunger was moved to compress the mixture to the desired compaction pressure. After a specified time interval, the temperature and compaction pressure were reduced. The compacted material was then removed from the die cavity, machined to the proper shape, and installed in the separators. One of the completed, assembled bearings is shown in Figure 10.

TABLE 6

COMPACTED MATERIAL COMPOSITIONS

Compact Identification	Composition Material	Composition Weight (%)
4-54-2	MoS_2	80
	Mo	5
	Ta	15
AFSL-14	WS_2	52.94
	Co	11.76
	Ag	35.3
AFSL-15	MoS_2	80
	Ta	20

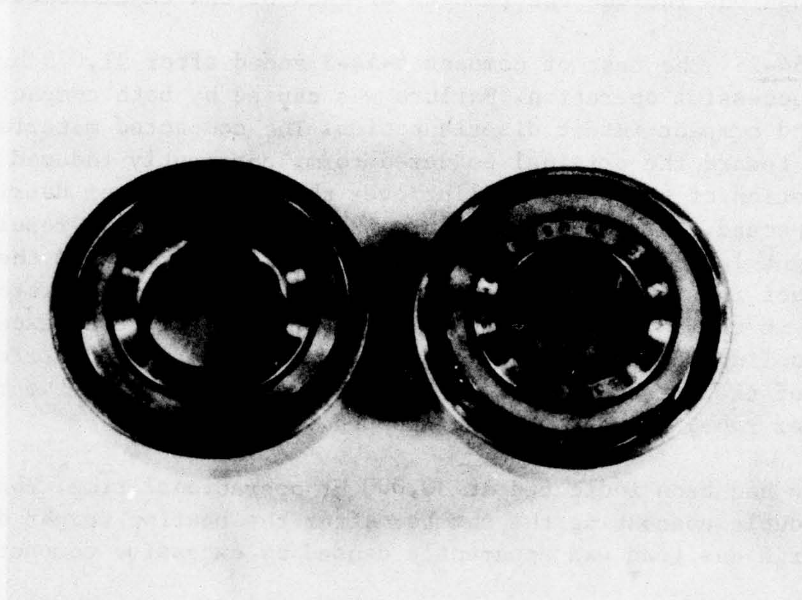

Figure 10 - Complete Bearing, with Compact Insert

Material properties such as friction coefficient, density, Poisson's ratio, and elastic moduli have been presented in Reference 12, in addition to information on wear rates. Other properties such as flexural strength and compressive strength were covered in Reference 16.

C. Experimental Results

Five tests were run on three different materials. Three of these experiments have been concluded; the other two continue to operate. The following discussions include the results of each of the experiments.

1. 4-54-2: The test of compact 4-54-2 ended after 31,075 hr (3.547 years) of successful operation. Failure was caused by both compact disintegration and compact insert disorientation. The compacted material appeared to progress toward the original powdered form, apparently induced in part by the vibration of the balls rolling over the previous wear debris. As might be expected, this mode of failure is self-exciting and resulted in extreme weight loss of compacted material from the bearing and the loosening of the compact inserts from the retainer. It was actually the rotation of the inserts in the retainer that caused the bearing torque to exceed the magnetic coupling torque. As the inserts rotated, the curved surface of the outer part of the inserts caused an interference fit between the retainer and the outer race, and the bearing stopped.

Trouble had been indicated at 30,000 hr operational time. The vacuum pump had trouble evacuating the chamber after the bearing weight data were taken. A large gas load was apparently caused by excessive compact powder formation.

The average wear rate for the first 28,000 hr was 4.05 μg/hr for 4-54-2. During the next 1,000 hr, the rate increased 7.5 times to 30.5 μg/hr. A routine inspection at 30,000 hr revealed a slight decrease in wear rate to 22.6 μg/hr. Another inspection was made at 30,376 hr, and it was discovered that the wear rate had more than doubled to 60.64 μg/hr. The bearing seized 699 hr later, during which time the average wear rate jumped to 318.5 μg/hr, 78.7 times what it was during the first 28,000 hr of "normal" operation. The weight-loss history is shown graphically in Figure 11. The weight of the bearing before the experiment began was 104.0533 g.

There are four apparent discrepancies in the resulting plot of weight loss versus time (Figure 11)--two weight gains (the time intervals were 200 to 500 hr and 17,000 to 18,000 hr) and two no-weight changes (11,000 to 12,000 hr and 23,000 to 24,000 hr). The calendar times of these discrepancies coincide with the periodic calibration of the scale used in this work. The large discrepancy (-0.0258 g) at the 200- to 500-hr interval was due to inaccurate recalibration. The scales were corrected later after another error was found.

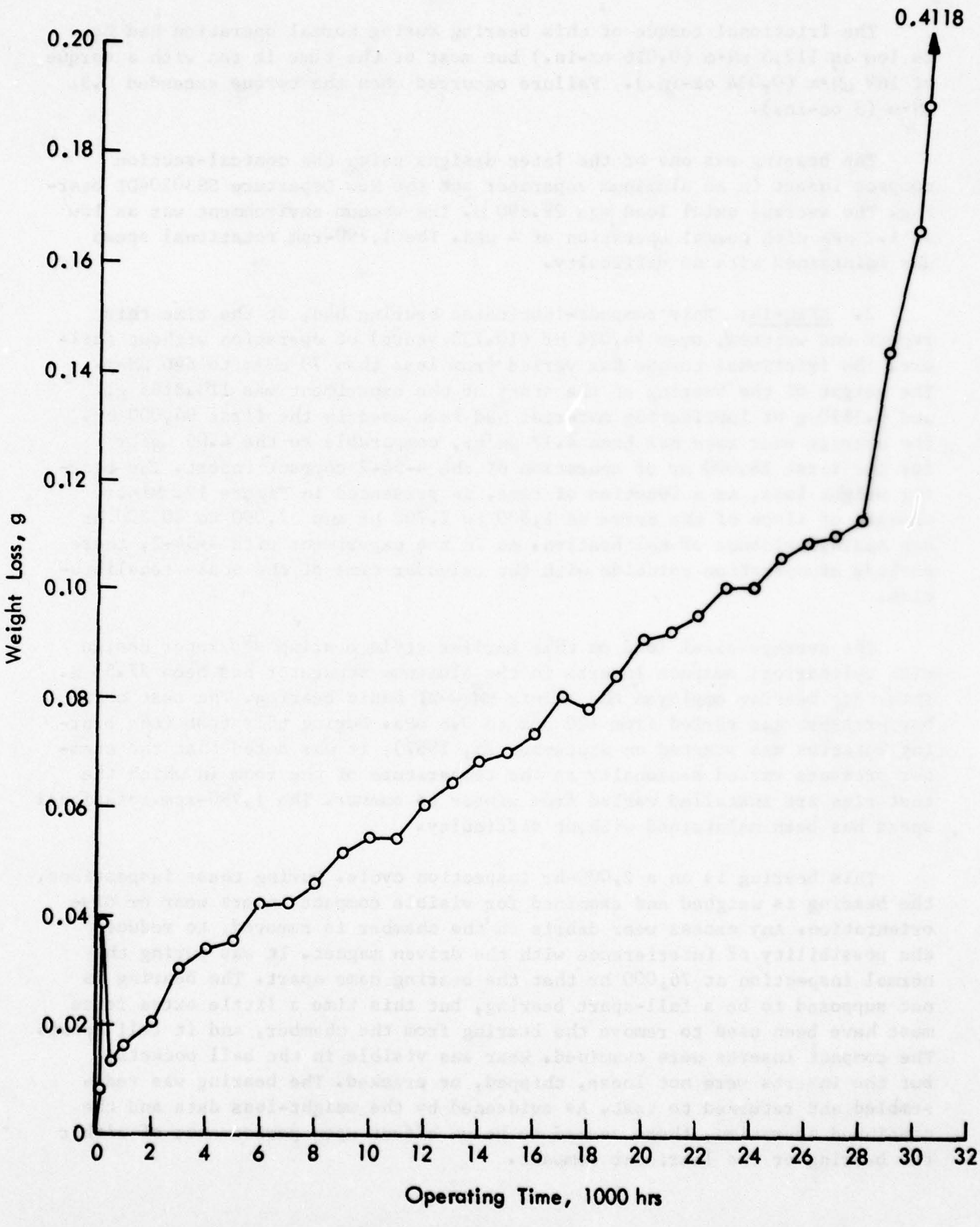

Figure 11 - Weight Loss as a Function of Time, 4-54-2

The frictional torque of this bearing during normal operation had been as low as 112.8 μN•m (0.016 oz-in.) but most of the time it ran with a torque of 169 μN•m (0.024 oz-in.). Failure occurred when the torque exceeded 0.31 μN•m (3 oz-in.).

The bearing was one of the later designs using the conical-section compact insert in an aluminum separator and the New Departure SS30204DT bearing. The average axial load was 29.890 N. The vacuum environment was as low as 1.2 μPa with normal operation of 4 μPa. The 1,790-rpm rotational speed was maintained with no difficulty.

2. AFSL-14: This compact-lubricated bearing had, at the time this report was written, over 94,024 hr (10.733 years) of operation without failure. The frictional torque has varied from less than 70 μN•m to 846 μN•m. The weight of the bearing at the start of the experiment was 113.8185 g, and 0.3870 g of lubricating material had been used in the first 94,000 hr. The average wear rate has been 4.12 μg/hr, comparable to the 4.05 μg/hr for the first 28,000 hr of operation of the 4-54-2 compact insert. The bearing weight loss, as a function of time, is presented in Figure 12. Minor changes of slope of the curve at 1,600 to 1,700 hr and 37,000 to 40,000 hr are again, evidence of calibration. As in the experiment with 4-54-2, these periods of operation coincide with the calendar time of the scale recalibration.

The average axial load on this earlier style bearing separator design with cylindrical compact inserts in the aluminum separator has been 27.53 N. This test bearing employed the Fafnir MM204WI basic bearing. The test chamber pressure has varied from 120 μPa to 0.8 μPa. During this test (the bearing rotation was started on September 25, 1967), it was noted that the chamber pressure varied seasonally as the temperature of the room in which the test rigs are installed varied from winter to summer. The 1,790-rpm rotational speed has been maintained without difficulty.

This bearing is on a 2,000-hr inspection cycle. During these inspections, the bearing is weighed and examined for visible compact insert wear or disorientation. Any excess wear debris in the chamber is removed, to reduce the possibility of interference with the driven magnet. It was during the normal inspection at 76,000 hr that the bearing came apart. The bearing is not supposed to be a fall-apart bearing, but this time a little extra force must have been used to remove the bearing from the chamber, and it fell apart. The compact inserts were examined. Wear was visible in the ball pockets, but the inserts were not loose, chipped, or cracked. The bearing was reassembled and returned to test. As evidenced by the weight-loss data and the continued operation, there seemed to be no effect upon performance of either the bearing or the lubricant compact.

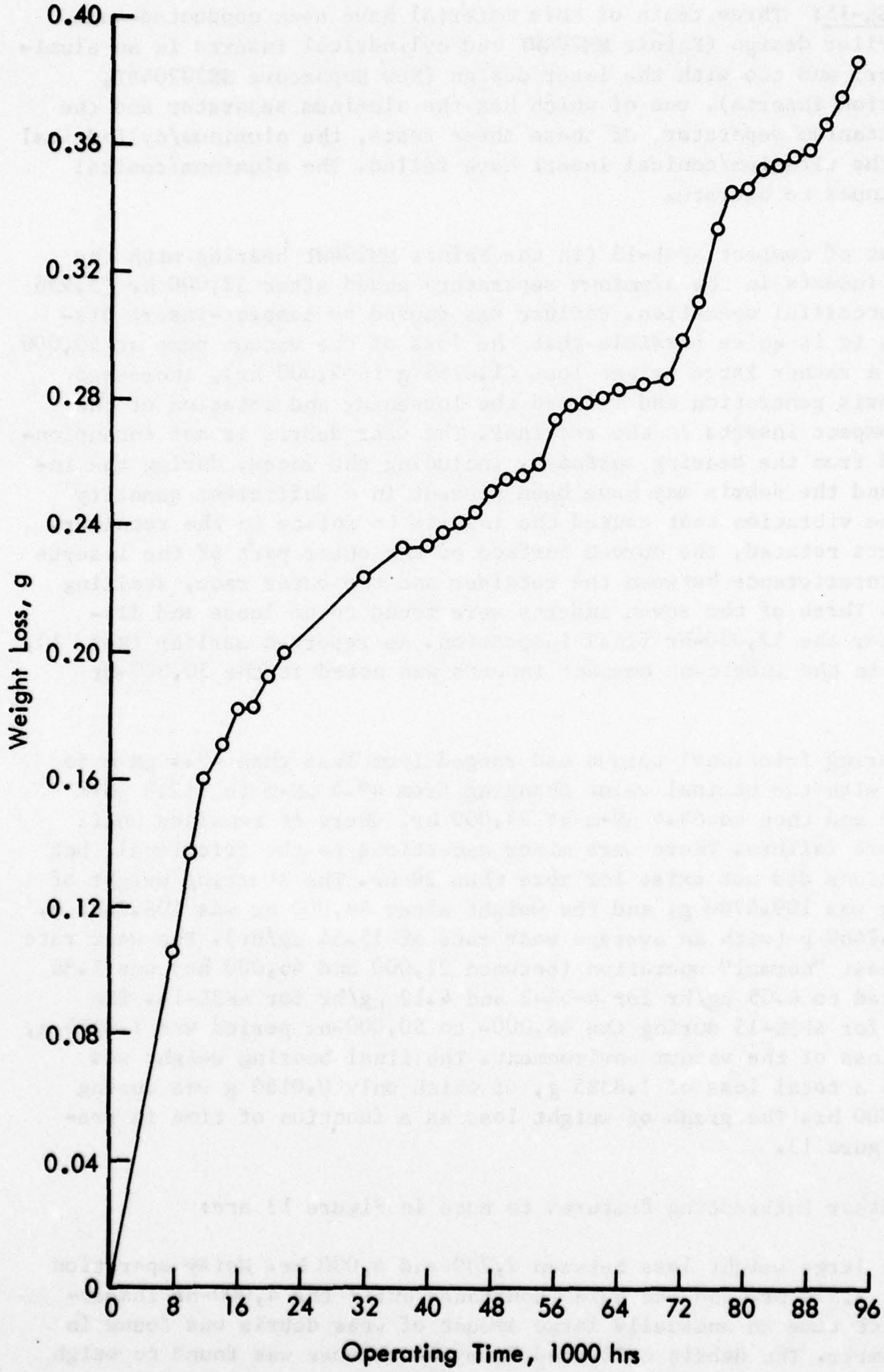

Figure 12 - Weight Loss as a Function of Time, AFSL-14

3. AFSL-15: Three tests of this material have been conducted--one with the earlier design (Fafnir MM204WI and cylindrical inserts in an aluminum separator) and two with the later design (New Departure SS30204DT, conical-section inserts), one of which has the aluminum separator and the other the titanium separator. Of these three tests, the aluminum/cylindrical insert and the titanium/conical insert have failed. The aluminum/conical insert continues to operate.

The test of compact AFSL-15 (in the Fafnir MM204WI bearing with the cylindrical inserts in the aluminum separator) ended after 52,000 hr (5.936 years) of successful operation. Failure was caused by compact-insert disorientation. It is quite possible that the loss of the vacuum pump at 50,000 hr, causing a rather large weight loss (1.0765 g in 2,000 hr), increased the wear debris generation and induced the loosening and rotation of the lubricant compact inserts in the retainer. The wear debris is not intentionally removed from the bearing surfaces, including the races, during the inspections, and the debris may have been present in a sufficient quantity to induce the vibration that caused the inserts to rotate in the retainer. As the inserts rotated, the curved surface of the outer part of the inserts caused the interference between the retainer and the outer race, stalling the bearing. Three of the seven inserts were found to be loose and disoriented after the 52,000-hr final inspection. As reported earlier (Ref. 1), a looseness in the lubricant compact inserts was noted at the 50,000-hr inspection.

The bearing frictional torque had ranged from less than 49.4 μN•m to 366.6 μN•m, with the nominal value changing from 49.4 μN•m to 112.8 μN•m at 16,000 hr and then to 63.4 μN•m at 23,000 hr, where it remained until shortly before failure. There were minor exceptions to the frictional, but these exceptions did not exist for more than 20 hr. The starting weight of this bearing was 109.4740 g, and the weight after 48,000 hr was 108.7485 g, a loss of 0.7460 g (with an average wear rate of 15.54 μg/hr). The wear rate during the last "normal" operation (between 21,000 and 46,000 hr) was 1.86 μg/hr compared to 4.05 μg/hr for 4-54-2 and 4.12 μg/hr for AFSL-14. The weight loss for AFSL-15 during the 48,000- to 50,000-hr period was 1.0765 g, due to the loss of the vacuum environment. The final bearing weight was 107.63555 g, a total loss of 1.8385 g, of which only 0.0160 g was during the last 2,000 hr. The graph of weight loss as a function of time is presented in Figure 13.

Three other interesting features to note in Figure 13 are:

1. The large weight loss between 2,200 and 4,000 hr. Noisy operation was noted at 3,376 hr, and the noise continued until the 4,000-hr inspection, at which time an unusually large amount of wear debris was found in the test chamber. The debris collected from the chamber was found to weigh 0.3197 g, accounting for a large percentage of the recorded weight loss for the period of operation (0.4033 g).

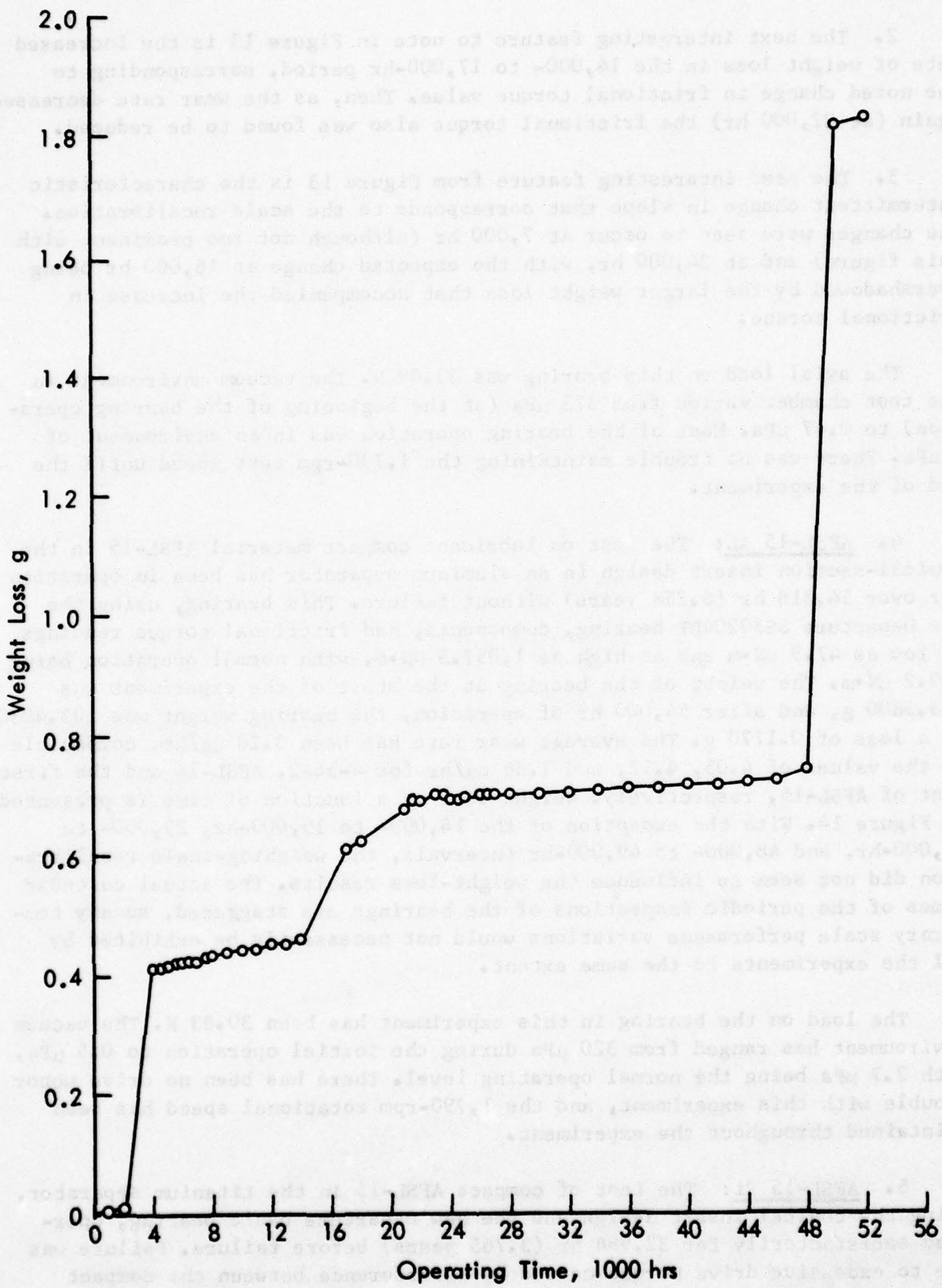

Figure 13 - Weight Loss as a Function of Time, AFSL-15

2. The next interesting feature to note in Figure 13 is the increased rate of weight loss in the 14,000- to 17,000-hr period, corresponding to the noted change in frictional torque value. Then, as the wear rate decreased again (at 22,000 hr) the frictional torque also was found to be reduced.

3. The next interesting feature from Figure 13 is the characteristic intermittent change in slope that corresponds to the scale recalibration. The changes were seen to occur at 7,000 hr (although not too prominent with this figure) and at 24,000 hr, with the expected change at 16,000 hr being overshadowed by the larger weight loss that accompanied the increase in frictional torque.

The axial load on this bearing was 33.09 N. The vacuum environment in the test chamber varied from 373 μPa (at the beginning of the bearing operation) to 0.47 μPa. Most of the bearing operation was in an environment of 8 μPa. There was no trouble maintaining the 1,790-rpm test speed until the end of the experiment.

4. AFSL-15 AL: The test on lubricant compact material AFSL-15 in the conical-section insert design in an aluminum separator has been in operation for over 54,818 hr (6.258 years) without failure. This bearing, using the New Departure SS30204DT bearing, components, had frictional torque readings as low as 47.9 μN•m and as high as 1,057.5 μN•m, with normal operation being 169.2 μN•m. The weight of the bearing at the start of the experiment was 103.5800 g, and after 54,000 hr of operation, the bearing weight was 103.4030 g, a loss of 0.1770 g. The average wear rate has been 3.28 μg/hr, comparable to the values of 4.05, 4.12, and 1.86 μg/hr for 4-54-2, AFSL-14 and the first test of AFSL-15, respectively. Weight loss as a function of time is presented in Figure 14. With the exception of the 14,000- to 15,000-hr, 23,000- to 25,000-hr, and 48,000- to 49,000-hr intervals, the weighing-scale recalibration did not seem to influence the weight-loss results. The actual calendar times of the periodic inspections of the bearings are staggered, so any temporary scale performance variations would not necessarily be exhibited by all the experiments to the same extent.

The load on the bearing in this experiment has been 30.83 N. The vacuum environment has ranged from 320 μPa during the initial operation to 0.5 μPa, with 2.7 μPa being the normal operating level. There has been no drive motor trouble with this experiment, and the 1,790-rpm rotational speed has been maintained throughout the experiment.

5. AFSL-15 Ti: The test of compact AFSL-15 in the titanium separator, using the conical insert design and the New Departure basic bearing, operated satisfactorily for 32,984 hr (3.765 years) before failure. Failure was due to excessive drive torque caused by interference between the compact insert and the outer race. The inspection at 32,000 hr revealed that one insert had turned in the separator and that the compact had experienced some slippage. Again, it appears that the wear debris in the ball raceway caused

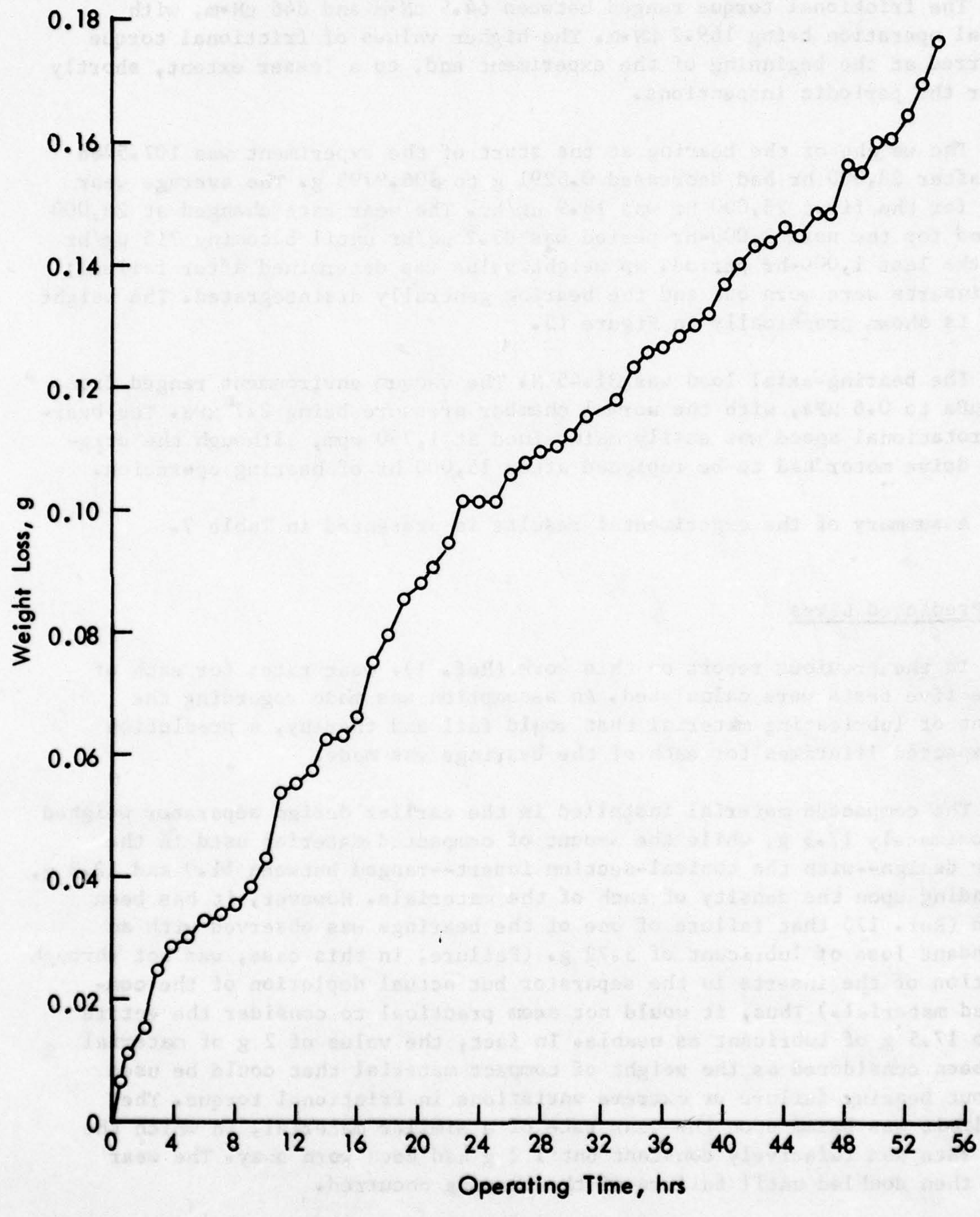

Figure 14 - Weight Loss as a Function of Time, AFSL-15 Al

excessive vibration, loosening and rotating the inserts until they rubbed the outer race.

The frictional torque ranged between 64.9 μN•m and 846 μN•m, with normal operation being 169.2 μN•m. The higher values of frictional torque occurred at the beginning of the experiment and, to a lesser extent, shortly after the periodic inspections.

The weight of the bearing at the start of the experiment was 107.5086 and after 28,000 hr had decreased 0.5291 g to 106.9795 g. The average wear rate for the first 28,000 hr was 18.9 μg/hr. The wear rate changed at 28,000 hr and for the next 3,000-hr period was 85.2 μg/hr until becoming 715 μg/hr for the last 1,000-hr period. No weight value was determined after failure; the inserts were worn out and the bearing generally disintegrated. The weight loss is shown graphically in Figure 15.

The bearing axial load was 31.45 N. The vacuum environment ranged from 332 μPa to 0.6 μPa, with the normal chamber pressure being 2.7 μPa. The bearing rotational speed was easily maintained at 1,790 rpm, although the original drive motor had to be replaced after 15,000 hr of bearing operation.

A summary of the experimental results is presented in Table 7.

D. Predicted Lives

In the previous report on this work (Ref. 1), wear rates for each of these five tests were calculated. An assumption was made regarding the amount of lubricating material that would fail and thereby, a prediction of expected lifetimes for each of the bearings was made.

The compacted material installed in the earlier design separator weighed approximately 17.5 g, while the amount of compacted material used in the later design--with the conical-section insert--ranged between 11.0 and 12.3 g, depending upon the density of each of the materials. However, it has been shown (Ref. 17) that failure of one of the bearings was observed with an attendant loss of lubricant of 3.72 g. (Failure, in this case, was not through rotation of the inserts in the separator but actual depletion of the compacted material.) Thus, it would not seem practical to consider the entire 11 to 17.5 g of lubricant as usable. In fact, the value of 2 g of material had been considered as the weight of compact material that could be used without bearing failure or extreme variations in frictional torque. The 2-g limit was based upon the wear rate of a similar material, in which the wear rate was relatively constant until 2 g had been worn away. The wear rate then doubled until failure of the bearing occurred.

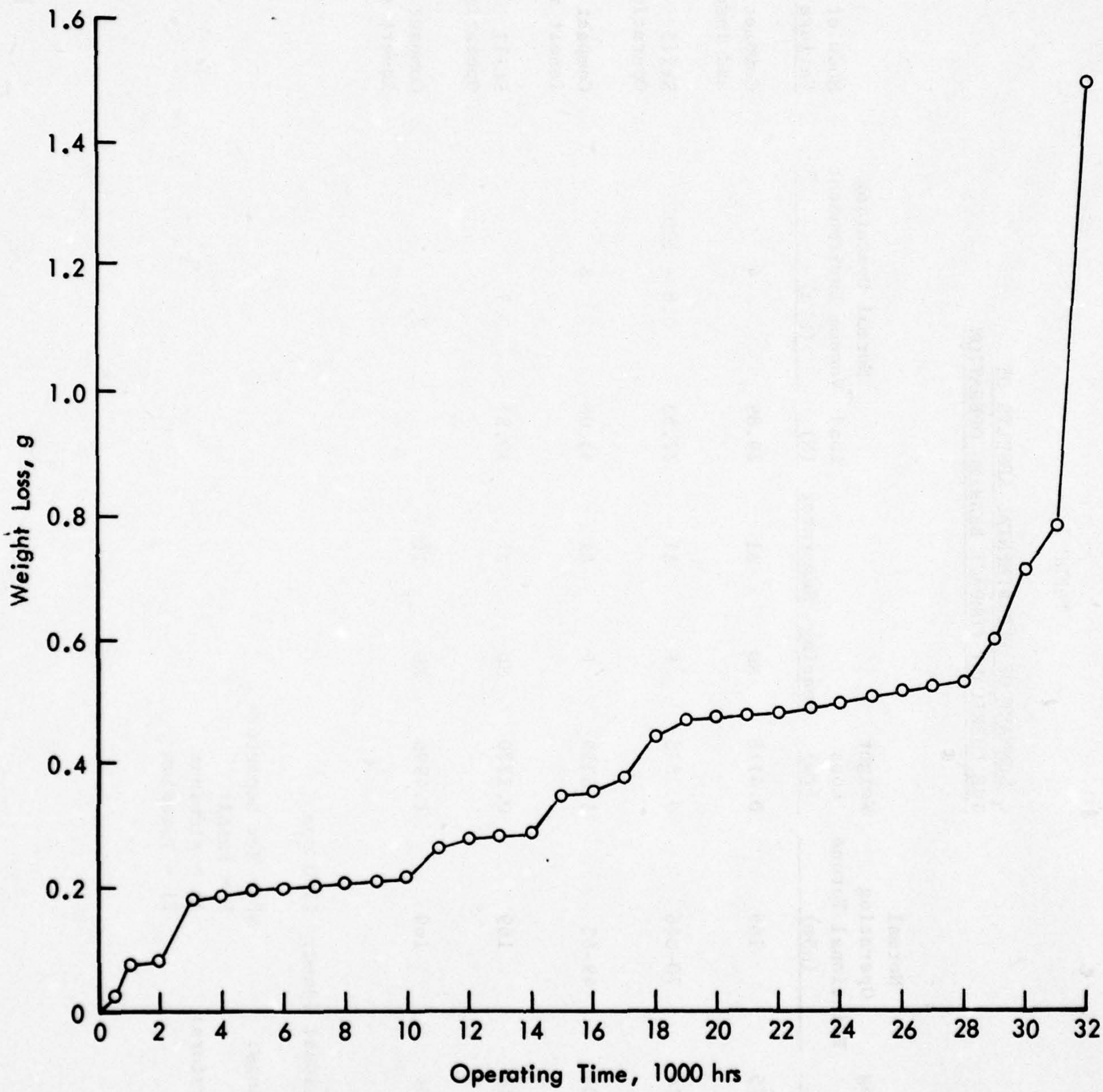

Figure 15 - Weight Loss as a Function of Time, AFSL-15 Ti

TABLE 7

SUMMARY OF EXPERIMENTAL RESULTS OF THE LUBRICANT COMPACT BEARING OPERATION

Material	Bearing Life (hr)	Normal Operating Frictional Torque (μNm)	Weight Loss (gm)	Bearing	Separator	Load (N)	Normal Operating Vacuum Environment (μPa)	Mode of Failure
4-54-2	31,075	169	0.4118	ND	Al	29.89	4	Compact disintegration and insert rotation
AFSL-14	94,024	70-846	0.3870	F	Al	27.53	0.8 - 120	Still Operating
AFSL-15	52,000	49-63	1.8385	F	Al	33.08	8	Compact insert rotation
AFSL-15	54,818	169	0.1770	ND	Al	30.83	2.7	Still Operating
AFSL-15	32-984	169	1.4996	ND	Ti		2.7	Compact insert rotation

Note: A. Rotational Speed: 1790 rpm

B. Bearings: ND = New Departure
F = Fafnir

C. Separators: Al = Aluminum
Ti = Titanium

As can be seen in Figures 11, 13, and 15, failures of these three bearings occurred well before 2 g of lubricant was used. In fact, the "liner" portion of these graphs for the "normal" operation seems to end at 0.11133 g, 0.7255 g, and 0.5291 g, respectively, for 4-54-2, AFSL-15, and AFSL-15 Ti. The other two tests that are still operating have used or lost 0.3870 g and 0.1770 g.

Although the life predictions (Ref. 1) were for many years more than those actually obtained with the three failed bearings, it must be noted that the failures were not due to depletion of the lubricant. Failure in each case was due to the failure of the bond between the compact insert and the retainer. Therefore, the prediction of wear-lives for the remaining bearings must use the additional assumption that the inserts will not rotate within the separator.

With the additional assumption that the usuable lubricant will be limited to 0.5 g of material, the following life predictions can be made.

For AFSL-14, the average wear rate has been 4.12 μg/hr for the first 28,000 hr of operation. Using 0.5 g of material, an operating life of 121,000 hr (13.8 years) should be expected. Thus, the test may have run for an additional 10.6 years, if the lubricant inserts had not rotated within the separator.

For AFSL-15 Al, the average wear rate has been 3.28 μg/hr for the first 54,000 hr of operation. Using 0.5 g of material, an operating life of 152,000 hr (17.4 years) should be expected. The test could run for an additional 11.2 years if the inserts do not rotate within the separator.

E. <u>Conclusions</u>

1. Long operating life can be obtained for bearings operating in a vacuum environment without the use of oil or grease lubricants. Two bearings have been operated for over 54,000 hr each without failure. Three other bearings operated satisfactorily for at least 31,000 hr before failing. One of the two bearings still in service has over 94,000 hr (10.73 years) of operation at 1,790 rpm using a lubricant compact material composed of tungsten disulfide, cobalt, and silver.

2. The cause of failure of the three bearings that stopped operating was not lubricant failure. The lubricant compact would have provided considerably longer operating lives if the bonding between the compact insert and the bearing retainer had not failed. The design of the bearing retainer should be modified to key the compact material in place and/or to improve the bonding technique to prevent lubricant compact rotation within the separator.

3. Weight loss from the bearings has been found to be a measurement criteria for lubricant compact performance. Weight loss has also shown to be indicative of operating life of the test specimens. Excessive loss also means reduced life of the test specimen due to depletion of the lubricant reserve.

4. Wear rate (loss of weight per unit time) has been calculated and used to predict the operating lives of the two bearings that are still rotating. These predictions reveal that wear-lives of 13 to 17 years are possible. These predictions are also made on the basis that the compact inserts do not become disoriented in their separators.

5. Operating frictional torque of all these bearings has been low, with a nominal value of 170 μN•m. There has been some variation of the frictional torque level, which for most applications would not be noticed. The low value of torque has been influenced by the requirement that each of the materials used in this work has been selected for its low friction coefficient under several operating conditions.

6. There can be no distinction made between the performance of the basic bearings. Both the Fafnir (made from 52100 steel) and the New Departure (made from 440C stainless steel) performed equally well, although the Fafnir bearing, started earlier, has more operating hours.

SECTION VIII

FRICTION AND WEAR OF SPUTTERED COATINGS ON GAS-BEARING COUPONS

Sputtered coatings of various materials have been used for obtaining special surface effects. These special surface effects can sometimes be obtained by adding a thin coating, 25 to 30 nm (250 to 500 Å) thick, to a surface. Sputtering is one method of applying these thin coatings. Sputtered coatings also provide good adherence to the substrate materials, good uniformity of thickness, high coating density, and the ability to readily coat curved and complex shapes. Based on these advantages, sputtering was used as a method to help solve a problem in the gyro bearings of a missile system.

The gyro bearings were hemispherically ended and ran as gas bearings. The problems experienced were with wear debris generated during start-up and coast-down when the mating surfaces rubbed. The sputtered coatings were to reduce the wear debris generation without increasing the frictional forces and still provide adequate rubbing life.

This part of the reported work deals with the determination of the friction coefficients and the wear-lives of various materials in different thicknesses and in various combinations.

A. Test Configuration and Apparatus

The test configuration for tests reported herein was a rigidly held ball on flat coupons. The coupons were 11.44 nm (0.4505 in.) in diameter and 2.54 nm (0.100 in.) thick. The material of construction was I-400 beryllium, the same as used in the gas bearings. A 3.18-nm (0.125-in.) diameter ball was the stationary, restrained member and the beryllium disc was the rotating member. After cleaning the ball-rider in acetone and drying it in a desiccator, it was placed in the ball-collet of the holding arm (see Figure 16). The holding arm, with its locating cross-pin, was placed in the stub end of the counter-balanced pivot arm. The two-piece arm design allows the ball, collet, and holding arm to be removed as an assembly from the test apparatus. The ball can be repositioned on the wear track at the same spot on the ball where contact was made before removal.

Frictional force was sensed by the displacement of the core of an LVDT (linear variable displacement transformer). The transducer core was supported on a shaft suspended between two leaf-springs that supplied the friction-resisting force. The shaft was connected to the ball-holding arm through a universally-mounted connecting link. When the ball-holding arm and connecting link were attached and ready for test, two relationships were established: (a) the fully assembled, unloaded arm was counter-balanced so that the arm

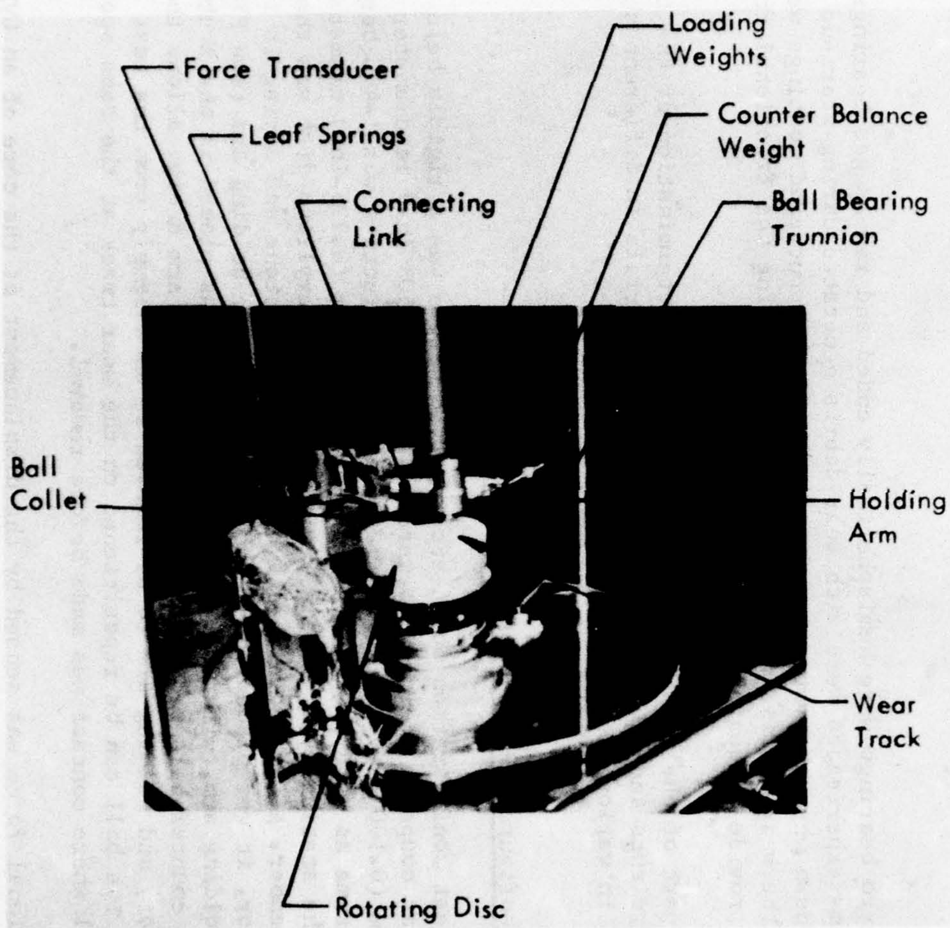

Figure 16 - Spatial Relationship of Test Zone

(mounted in a ball bearing trunnion) was level; and (b) the ball was placed at a distance of 5 mm from the center of rotation of the coupon. When the coupon was rotated at 60 rpm, the point-of-contact zone passed beneath the spherical rider at 31.42 mm/sec. The spatial relationships of the various components are shown in Figure 16.

The bearing coupon was attached to the driving head and rotation was started. After the speed of 60 rpm was established, the deadweight load was added to the ball collet and the ball-rider was allowed to contact the disc.

Frictional force was recorded on a strip-chart recorder and the operating time was determined. The tests were considered to have failed when the frictional force had increased to 1.25 times the operating frictional force level. Sometimes, the failure was so rapid that the last frictional force recorded exceeded the operational level by more than 1.25 times the operating value.

During some of the tests, a stream of dry nitrogen gas was directed at the contact zone to provide a cover gas for the operation.

B. Test Results

The results of the 163 tests are presented in Table 8. The information regarding the lubricants will be further explained in this section. Due to the nature of the tests, with the different types of lubricants, different loads, different ball material, and the use of nitrogen as a cover gas in some of the experiments, the specimen number will be used in the discussion of the test results. The specimen numbers are not always in numerical order, since the numbers were assigned only as a function of the time of receipt of the specimen.

Two loads were used: 15 g and 5 g. All the testing started out with 15 g, but with some of the early failures, it was not possible to see the effects of the lubricant-coating processing parameters. Therefore, the loads were reduced to 5 g for some tests to see if processing parameter effects could be determined. With the 15-g load on the beryllium coupon, the maximum Hertz contact stress was calculated to be 745 MPa (108, 126 psi), using the formula for Case 1 from Reference 18. With the 5-g load, the Hertz stress was 517 MPa (74,970 psi). The assumption has been made that the thin film does not influence the stress level. The author was unable to find which would encompass the effects of thin contaminating layers between the two bodies in a concentrated contact (Hertzian).

TABLE 8

RESULTS OF FRICTION AND WEAR-LIFE TESTS ON GAS-BEARING COUPONS

Lubricant		Specimen Number	Load (g)	Ball Material	Cover Gas	Friction Coefficient Start	Friction Coefficient Run	Friction Coefficient Finish	Operating Time (min)
TiC	1 μin.	1	5	52100	N_2	0.233–0.253	0.227–0.243	0.313	36.1
	dc	2	5	52100	N_2	0.200–0.217	0.193–0.220	0.287	12.6
		3	5	52100	N_2	0.173–0.183	0.173–0.187	0.267	12.9
		4	15	52100	N_2	0.080–0.087	0.058–0.063	0.089	2.2
		5	15	52100	N_2	0.093–0.104		0.136	3.9
TiC	5 μin.	11	5	52100	N_2	0.107–0.117	0.120–0.133	0.243	26.9
	dc	12	5	52100	N_2	0.133–0.150	0.133–0.147	0.317	23.8
		13	5	52100	N_2	0.107–0.140	0.107–0.140	0.280	14.6
		14	15	52100	N_2	0.058–0.063		0.116	3.0
		15	15	52100	N_2	0.062–0.069		0.112	2.3
TiC	5 μin.	16	15	TiC	Air	0.256	0.267	0.427	36.5
MoS_2	15 μin.	17	15	TiC	Air	0.224–0.235	0.245–0.277	0.352	4.9
	dc	18	15	TiC	Air	0.235–0.251	0.256–0.341	0.389	2.6
		19	15	TiC	Air	0.245–0.267		0.331	4.0
		20	5	52100	Air	0.173–0.227	0.177–0.187	0.284	21.2
TiC	5 μin.	21	15	TiC	Air	0.061–0.068		0.105	4.1
MoS_2	15 μin.	22	15	TiC	Air	0.059–0.067	0.072–0.077	0.096	5.4
	rf	23	15	TiC	Air	0.109–0.125	0.123–0.160	0.235	3.0
		24	15	TiC	Air	0.112–0.116		0.128	0.4
		25	5	52100	Air	0.307–0.560		0.792	0.8

TABLE 8 (continued)

Lubricant		Specimen Number	Load (g)	Ball Material	Cover Gas	Friction Coefficient Start	Run	Finish	Operating Time (min)
TiC	5 μin.	26	15	TiC	Air	0.093-0.109	0.083-0.088	0.141	3.7
MoS_2+	15 μin.	27	15	TiC	Air	0.117-0.171	0.057-0.080	0.219	4.5
Sb_2O_3	dc	28	15	TiC	Air	0.107-0.131	0.085-0.107	0.267	0.7
		29	15	TiC	Air	0.128-0.149	0.107-0.117	0.163	20.0
		30	5	52100	Air	0.613-0.800	0.760-0.800	1.413	0.1
TiC	5 μin.	31	15	TiC	Air	0.085	0.061	0.179	0.5
MoS_2+	15 μin.	32	15	TiC	Air	0.064-0.077	0.035-0.043	0.072	26.4
Sb_2O_3	rf	33	15	TiC	Air	0.080-0.107	0.048-0.059	0.148	36.9
		34	15	TiC	Air	0.069-0.077	0.059-0.063	0.085	7.5
		35	5	52100	Air	0.373-0.453		0.747	0.3
TiC	5 μin.	70[a]/	5	52100	Air	0.387-0.467	0.333-0.427	0.752	280.6
AD13	1 μin.	71	5	52100	Air	0.277-0.297		0.535	0.3
		72	5	52100	N_2	0.233-0.247		0.333	4.0
		74	5	52100	N_2	0.240-0.270		0.357	2.8
		73	15	52100	Air	0.222-0.245		0.367	0.6
TiC	5 μin.	75	5	52100	Air	0.293-0.310		0.393	0.7
AD13	5 μin.	76	5	52100	Air	0.287-0.300		0.509	1.1
		77	5	52100	N_2	0.280-0.317		0.517	0.7
		79	5	52100	N_2	0.107-0.153		0.347	0.3
		78[a]/	15	52100	Air	0.516-0.533		0.556	8.6
TiC	5 μin.	80	5	52100	Air	0.287-0.300		0.480	1.9
AD13	15 μin.	81	5	52100	Air	0.420-0.447		0.600	0.4
		82	5	52100	N_2	0.360-0.380		0.477	2.2
		84	5	52100	N_2	0.373-0.403		0.637	1.1
		83[a]/	15	52100	Air	0.516-0.533	0.302-0.329	0.587	4.3

TABLE 8 (continued)

Lubricant		Specimen Number	Load (g)	Ball Material	Cover Gas	Friction Coefficient Start	Friction Coefficient Run	Friction Coefficient Finish	Operating Time (min)
TiC	5 μin.	85	5	52100	Air	0.153-0.170		0.473	0.9
AD14	1 μin.	86	5	52100	Air	0.220-0.333		0.557	0.7
		87	5	52100	N_2	0.157-0.173		0.283	0.4
		89	5	52100	N_2	0.153-0.173		0.303	0.5
		88	15	52100	Air	0.271-0.347		0.572	0.8
TiC	5 μ	90	5	52100	Air	0.137-0.157		0.373	1.0
AD14	5 μ	91	5	52100	Air	0.190-0.203		0.363	0.9
		92	5	52100	N_2	0.240		0.640	0.0
		94	5	52100	N_2	0.267-0.620		0.693	0.1
		93	15	52100	Air	0.160-0.200		0.320	0.7
TiC	5 μ	95	5	52100	Air	0.200		0.587	0.9
AD14	15 μ	96	5	52100	Air	0.183-0.210	0.123-0.147	0.353	18.2
		97	5	52100	N_2	0.433		0.603	0.0
		99	5	52100	N_2	0.103-0.360		0.597	0.3
		98	15	52100	Air	0.320-0.409		0.649	0.3
TiC	Thin	133	15	52100	Air	0.498-0.542		0.671	0.8
MoS_2	Low Bias	134	15	52100	Air	0.364-0.560	0.160-0.222	0.356	3.3
		135	15	52100	Air	0.444-0.560		0.684	0.7
TiC	Thin	136	15	52100	Air	0.422-0.551	0.302-0.436	0.609	1.6
MoS_2	High Bias	137	15	52100	Air	0.320-0.453		0.569	0.4
		138	15	52100	Air	0.427-0.604	0.427-0.489	0.613	1.1
TiC	Thick	139	15	52100	Air	0.462-0.560	0.204-0.373	0.538	3.6
MoS_2	Low Bias	140	15	52100	Air	0.489-0.569		0.662	1.2
		141	15	52100	Air	0.356-0.418	0.258-0.311	0.476	1.4

TABLE 8 (continued)

Lubricant		Specimen Number	Load (g)	Ball Material	Cover Gas	Friction Coefficient Start	Run	Finish	Operating Time (min)
TiC	Thick	142	15	52100	Air	0.204-0.253	0.089-0.160	0.276	1.4
MoS_2	High Bias	143	15	52100	Air	0.222-0.320	0.053-0.071	0.151	4.0
		144	15	52100	Air	0.498-0.622	0.418-0.533	0.702	1.7
MoS_2	Thin	145	15	52100	Air	0.440-0.538		0.698	0.5
	Low Bias	146	15	52100	Air	0.382-0.409		0.667	0.3
		147	15	52100	Air	0.311-0.382		0.653	0.3
MoS_2	Thin	148	15	52100	Air	0.427-0.493		0.658	0.7
	High Bias	149	15	52100	Air	0.222-0.347		0.578	0.7
		150	15	52100	Air	0.316-0.444		0.596	0.3
MoS_2	Thick	151	15	52100	Air	0.293-0.400	0.089-0.178	0.271	6.3
	Low Bias	152	15	52100	Air	0.324-0.391	0.133-0.196	0.347	4.7
		153	15	52100	Air	0.347-0.427	0.267-0.311	0.422	0.3
MoS_2	Thick	154	15	52100	Air	0.116-0.151	0.124-0.142	0.258	56.8
	High Bias	155	15	52100	Air	0.258-0.373	0.089-0.160	0.293	5.5
		156	15	52100	Air	0.404-0.538		0.649	0.2
MoS_2	15 μin.	110[b]	15	TiC	Air	0.142-0.258	0.053-0.089	0.187	
Sb_2O_3	rf	111[b]	15	TiC	Air	0.188-0.280	0.062-0.076	0.187	
		112	15	TiC	Air	0.187-0.206		0.316	0.4
		113	15	52100	Air	0.311-0.347		0.373	0.1
		114	15	52100	Air	0.222-0.267		0.298	0.1

TABLE 8 (continued)

Lubricant		Specimen Number	Load (g)	Ball Material	Cover Gas	Friction Coefficient Start	Run	Finish	Operating Time (min)
MoS_2	5 μin.	115	15	TiC	Air	0.093-0.133	0.111-0.133	0.207	74.1
Sb_2O_3	rf	116	15	TiC	Air	0.107-0.138	0.084-0.093	0.227	53.8
		117	15	TiC	Air	0.082-0.102	0.116-0.133	0.233	65.1
		118	15	52100	Air	0.124-0.169	0.102-0.116	0.249	15.7
		119	15	52100	Air	0.116-0.133	0.089-0.107	0.178	9.7
		130	15	52100	Air	0.178		0.311	0.1
		131	15	52100	Air	0.209-0.262	0.071-0.107	0.236	7.0
		132	15	52100	Air	0.196-0.462	0.025-0.133	0.240	4.4
MoS_2	5 μin.	120	15	TiC	Air	0.222-0.587		0.734	0.9
Sb_2O_3	dc	121	15	TiC	Air	0.231-0.462		0.523	0.6
		122	15	TiC	Air	0.267-0.507		0.693	0.5
		123	15	52100	Air	0.169-0.302		0.584	0.4
		124	15	52100	Air	0.276-0.324		0.667	0.4
MoS_2	15 μin.	125	15	TiC	Air			0.842	0
Sb_2O_3	dc	126	15	TiC	Air			0.768	0
		127	15	TiC	Air			0.827	0
		128	15	52100	Air	0.613		0.711	0.1
		129	15	52100	Air	0.462		0.698	0.2
WC	5 μin.	6	5	52100	N_2	0.133-0.170	0.127-0.143	0.195	12.7
	dc	7	5	52100	N_2	0.133-0.160		0.245	2.4
		8	5	52100	N_2	0.130-0.143		0.215	0.9
WC	1 μin.	36	5	52100	N_2	0.080-0.095	0.050-0.067	0.095	30.5
	dc	37	5	52100	N_2	0.093-0.121	0.080-0.107	0.152	11.0
		38	5	52100	N_2	0.083-0.093		0.141	2.6

TABLE 8 (continued)

Lubricant		Specimen Number	Load (g)	Ball Material	Cover Gas	Friction Coefficient Start	Run	Finish	Operating Time (min)
WC	5 μin.	52	15	TiC	Air	0.277–0.320		0.459	3.9
AD13	1 μin.	53	15	TiC	Air	0.178	0.178	0.240	2.8
		54	15	TiC	Air	0.196–0.204		0.284	1.0
WC	5 μin.	55	15	TiC	Air	0.356		0.604	0.1
AD13	5 μ	56	15	TiC	Air	0.356		0.613	0.2
		57	15	TiC	Air	0.311–0.356	0.311–0.364	0.708	0.7
WC	5 μin.	58	15	TiC	Air	0.395		0.640	0.1
AD13	15 μin.	59	15	TiC	Air	0.604	0.444	0.649	1.0
		60	15	TiC	Air	0.578–0.596		0.658	2.2
WC	5 μin.	61	15	TiC	Air	0.160–0.196	0.151–0.187	0.240	18.4
AD14	1 μin.	62	15	TiC	Air	0.133–0.178		0.213	1.0
		63	15	TiC	Air	0.169–0.222		0.287	1.1
WC	5 μin.	64	15	TiC	Air	0.142–0.231	0.151–0.196	0.240	0.1
AD14	5 μin.	65	15	TiC	Air	0.267	0.142–0.182	0.311	0.1
		66	15	TiC	Air	0.178–0.191		0.261	2.3
WC	5 μin.	67	15	TiC	Air	0.178	0.213–0.222	0.278	1.2
AD14	15 μin.	68	15	TiC	Air	0.178		0.240	0.1
		69	15	TiC	Air	0.231–0.249	0.124–0.160	0.191	16.8
Au	7–70 Å	157	5	52100	N_2	0.193–0.233		0.370	0.3
MoS_2	6–1 μin.	158	5	52100	N_2	0.223–0.287		0.373	1.5
Sb_2O_3		159	5	52100	N_2	0.280–0.313		0.473	0.2
		160	5	52100	N_2	0.229		0.367	0.3
		161	5	52100	N_2	0.187–0.193		0.335	0.1

TABLE 8 (continued)

Lubricant		Specimen Number	Load (g)	Ball Material	Cover Gas	Friction Coefficient Start	Run	Finish	Operating Time (min)
MoS_2	45°	162	5	52100	N_2	0.293-0.313		0.467	0.7
Sb_2O_3		163	5	52100	N_2	0.353-0.380		0.567	0.2
		164	5	52100	N_2	0.447-0.467		0.693	0.2
MoS_2	90°	165	5	52100	N_2	0.193-0.242		0.290	0.4
Sb_2O_3		166	5	52100	N_2	0.287		0.463	0.0
		167	5	52100	N_2	0.220		0.437	0.0
MoS_2	Intermittent	168	5	52100	N_2	0.080-0.113	0.013-0.033	0.127	9.2
Sb_2O_3		169	5	52100	N_2	0.043-0.057	0.027-0.047	0.100	4.5
		170	5	52100	N_2	0.080-0.137	0.020-0.040	0.093	1.9
MoS_2	Vented	171	5	52100	N_2	0.240-0.333	0.077-0.227	0.300	0.8
Sb_2O_3		172	5	52100	N_2	0.147-0.307		0.380	0.1
		173	5	52100	N_2	0.280		0.430	0.1
MoS_2		174	5	52100	N_2			0.653	0.0
Sb_2O_3		175	5	52100	N_2	0.353-0.393		0.513	0.7
Teflon		176	5	52100	N_2	0.307-0.353		0.573	0.3
MoS_2	Delayed	177	5	52100	N_2	0.233-0.247		0.580	0.1
Sb_2O_3		178	5	52100	N_2	0.320		0.657	0.3
		179	5	52100	N_2	0.387		0.673	0.2
MoS_2	Delayed	182	5	52100	N_2	0.333		0.661	0.0
Sb_2O_3	Ar + O_2	183	5	52100	N_2	0.207-0.220		0.523	0.2
		184	5	52100	N_2	0.133		0.693	0.1

TABLE 8 (concluded)

Lubricant		Specimen Number	Load (g)	Ball Material	Cover Gas	Friction Coefficient Start	Run	Finish	Operating Time (min)
MoS_2	Delayed	187	5	52100	N_2	0.343		0.545	0.1
Sb_2O_3	Ar + BCl_3	188	5	52100	N_2	0.487-0.513	0.447-0.480	0.607	3.4
		189	5	52100	N_2	0.460-0.520	0.387-0.407	0.553	3.0
MoS_2	Alternate	192	5	52100	N_2	0.220		0.427	0.3
Sb_2O_3	Deposition/	193	5	52100	N_2	0.207-0.253		0.587	0.3
	Oxidation	194	5	52100	N_2	0.160-0.207		0.540	0.4

a/ The friction trace was unusual. The normal trace has spikes appearing in the increasing friction direction. This trace had decreasing friction spikes, as if the wear was causing lubricant to be brought back into the wear track.

b/ These two tests had extremely erratic operation. Failure would normally have been listed as immediate, except that the tests were continued. Both had decreasing frictional forces (still erratic) until the lubricant seemed to be wiped from the contact zone. The "bare" metal operation was continued until the frictional force increased (when the MoS_2 was apparently wiped from the contact zone). "Failure" times were then 15.1 and 3.6 min for 110 and 111, respectively. Both traces had the same features, except for the "foreshortened" opeation of 111.

Note: 1 μin. = 25 nm
5 μin. = 127 nm
15 μin. = 381 nm
70 Å = 7 nm

If the frictional trace was steady, only one value of the frictional coefficient was listed in the table. If the frictional trace oscillated, the low and high values are reported and reflect the variation in frictional force. When a running friction coefficient is not given, the frictional trace showed only that the starting friction was the lowest obtained and that the frictional force increased until the failure criteria had been reached.

For specimens 1 through 5, the lubricant is listed as TiC, 1-in., dc. This means that the hard coating applied to the beryllium was titanium carbide, the coating was 25 nm thick, and the sputtering was done using direct current. The running friction coefficient averaged 0.207 for the three tests with a 5-g load. The average time for these three tests was 20.53 min. With this small sample-size, statistical spread was $\pm$ 13.48 min with a 68% probability of the speed containing the actual average. That is, with a 68% probability of being correct; the average wear-life for this material should be between 7.05 and 34.01 min. For the heavier load on the hard surface, the average friction coefficient dropped to 0.080 and the average life dropped to 3.05 min. Even with the lower frictional coefficient, the average frictional force was greater for the 15-g load (1.2-g) than for the 5-g load (1.0-g).

For specimens 11 through 15, the only change was in the thickness of the titanium carbide hard coating, ranging from 25 to 127 nm. The average friction coefficient was 0.130 for the three tests with 5-g loads. The average wear-life was 21.77 min, with an expected range from 15.37 to 28.17 min. Statistically speaking, there is no difference in wear-lives between the two thicknesses. The larger load of 15 g again reduced the average friction coefficient (to 0.063) and the average wear-life (to 2.65 min). The average frictional force for the heavier load (0.94 g) was still greater than the lighter load (0.65 g).

The lubricant listed for specimens 16 through 20 is a hard coat of titanium carbide, 127 nm thick, with a layer of MoS_2, 381 nm thick, both applied with dc sputtering. The average friction coefficient was 0.271 for the 15-g load condition. The ball-rider used for the heavier loads was titanium carbide, and no nitrogen cover gas was used. The average wear-life was 12.0 min, but due to the large scatter, any operating time up to 28.36 min would have been equally expected. Specimen 20 was tested with a 52100 steel ball in air. The average friction coefficient was 0.182 and the operating time was 21.2 min. The comparison between specimen 20 and specimens 1 through 3 and 11 through 13 shows that the presence of MoS_2 (or whatever was transferred) did not seem to affect either the friction coefficient or the wear-life results.

Specimens 21 through 25 had the same coatings applied as did specimens 16 through 20, with the exception that an rf (radio frequency) was used for the sputtering. The average friction coefficient for the 15-g load specimens

(21 through 24) was 0.099; the wear-life average was 3.22 min, with any time between 1.10 and 5.35 min having the same 68% probability of being correct. The average friction coefficient was approximately one-third the value for the dc sputtered coating. With the exception of specimen 16, the wear-lives are comparable, although too short to be of practical value. Apparently, the test with specimen 25 exhibited high friction and an extremely short operating time.

A synergistic lubricating effect has been found when MoS_2 and Sb_2O_3 (antimony trioxide) are mixed and used in sprayed, bonded film lubricants. Therefore, the two powders were mixed and sputtered together for specimens 26 through 30. The average friction coefficient for specimens 26 through 29 was 0.090 (one-third of that for specimens 16 through 19).

The average wear-life was 7.22 min, with any time up to 15.90 min being equally acceptable, due to the data scatter. Specimen 30 performed in a manner similar to specimen 25.

Specimens 31 through 35 differed from Specimens 26 through 30 only in the use of rf versus dc sputtering. The rf-sputtered MoS_2 had an average friction coefficient of 0.053 (60% of the value for the dc version of the same coating). Wear-life data scatter again influenced the average wear-life (17.82 min); any value between 1.05 and 34.60 min can be expected. The lightly loaded condition, with a 52100 steel ball-rider, still exhibits poor friction and wear results.

The next three sets of specimens (70 through 74, 75 through 79, and 80 through 84) differed only in the thickness of the lubricant identified only as AD13 by the supplier. For specimens 70 through 74, the thickness was 25 nm; for specimens 75 through 79, the thickness was 127 nm; and for specimens 80 through 84, the thickness was 381 nm. No effect on friction was detectable, with the comparisons being made between 0.334, 0.298, and 0.364 (specimens 70 through 71, 75 through 76, and 80 through 81); 0.248, 0.214, and 0.379 (specimens 72 and 74, 77 and 79, and 82 and 84); and 0.234, 0.524, and 0.316 (specimens 73, 78, and 83). The wear-life data regarding the effect of thickness of the top coat were inconclusive and too short to be practical values. Some of these tests (specimens 70, 78, and 83) had frictional traces that were unusual in the fact that the friction "spikes" were decreasing spikes (instead of the normally increasing spikes) as if the lubricant were being dragged back into the contact zone.

The next three sets of specimens (85 through 89, 90 through 94, and 95 through 99) were similar to the last three sets, except that the lubricant was AD14 instead of AD13. Again, no effect of thickness was detectable, with comparisons being made between 0.219, 0.172, and 0.157 (specimens 85 and 86, 90 and 91, and 95 and 96); 0.164, 0.376, and 0.299 (specimens 87 and 89, 92 and 94, and 97 and 99); and 0.309, 0.180, and 0.364 (specimens 88, 93, and 98). The wear-life data were also inconclusive regarding the effect of thickness of the top coat. One might hazard a guess and decide that AD14 was better than AD13 as a lubricant, based on average friction coefficients of 0.244 for AD14 compared to 0.323 for AD13.

The next eight groups have been considered as a subgroup. In these eight subgroups of three (specimens 133 through 156), four groups (specimens 133 through 144) were prepared with titanium carbide hard-coated surfaces covered with MoS_2, and the other four groups (specimens 145 through 156) were coated with MoS_2 only. This comparison revealed that the specimens coated with only MoS_2 had lower average friction than the specimens with MoS_2 over the TiC base. No conclusions could be reached on the wear-lives. Of these two groupings, each had half of the specimens prepared with thin coats and the other had thick coats. (The actual thicknesses were not specified by the supplier.) This comparison revealed that the thicker coats were better--lower average friction coefficient and longer wear-lives. Half of the specimens were prepared using a high bias voltage. The tests are shown in Table 9.

No conclusions could be reached regarding either friction coefficient, average friction coefficient and average wear-lives, and wear-life ranges or wear-life results.

The next groups have also been considered as a subgroup. In these four groups of five or eight (specimens 110 through 132), two groups (specimens 110 through 119 and 130 through 132) were prepared using rf sputtering, and two groups (specimens 120 through 129) were prepared using dc sputtering. This comparison revealed that rf was better, both for lower friction coefficients and longer wear-lives. Of these two groupings, approximately half were coated with thin (127-nm) coatings and the others had a 381-nm coating. For the conditions used, the thinner coats were better. Wear-life data were inconclusive. Again, approximately half were tested using a titanium carbide ball-rider and the other half were tested with a 52100 steel ball-rider. The 52100 ball gave marginally better friction results. Wear-life data again were inconclusive. The test results, average friction coefficients, and wear-lives are shown in Table 10.

The next two groups (specimens 6 through 8 and 36 through 38) were prepared using tungsten carbide as the hard coat and with no lubricant. The two groups were prepared with two different thicknesses: 127 nm (specimens 6 through 8) and 25 nm (specimens 36 through 38). This comparison revealed that the thinner coats, with an average friction coefficient of 0.080, had an average wear-life of 14.7 min, compared to the thicker coats, with an average friction coefficient of 0.139 and an average wear-life of 5.33 min. A comparison between specimens 6 through 8 and 11 through 13 can be made in which the tungsten carbide surfaces of specimens 6 through 8 had a higher friction coefficient (0.139) than the titanium carbide friction coefficient (0.130) and the average wear-life advantages also belonged to the titanium carbide. Another comparison between specimens 36 through 38 and 1 through 3

TABLE 9

AVERAGE FRICTION COEFFICIENTS AND WEAR-LIVES FOR SUBGROUP 133-156

Specimen Group	Coating	Thickness	Bias	Average Friction Coefficient	Average Wear-Life (min)	Wear-Life Range (min)
133-135	TiC/MoS_2	Thin	Low	0.404	1.60	0.13-3.07
136-138			High	0.404	1.03	0.43-1.63
139-141		Thick	Low	0.367	2.07	0.74-3.40
142-144			High	0.221	2.37	0.95-3.79
145-147	MoS_2	Thin	Low	0.410	0.37	0.25-0.49
148-150			High	0.375	0.57	0.34-0.80
151-153		Thick	Low	0.196	3.77	0.66-6.88
154-156			High	0.243	20.83	0-52.09

TABLE 10

AVERAGE FRICTION COEFFICIENTS AND WEAR-LIVES FOR SUBGROUP 110-132

Specimen Group	Sputtering	Thickness (nm)	Ball-Rider Material	Average Friction Coefficient	Average Wear-Life (min)
115-117	rf	127	TiC	0.112	64.33
118-119, 130-132			52100	0.103	7.38
110-112		381	TiC	0.112	0
113-114			52100	0.287	0.1
120-122	dc	127	TiC	0.379	0.67
123-124			52100	0.268	0.4
125-127		381	TiC	0.812	0
128-129			52100	0.538	0.15

can be made in which the friction results were reversed (tungsten carbide with 0.080 average and titanium carbide with 0.207 average), but the wear-life advantage still favored the titanium carbide.

The next three groups of three (specimens 52 through 60) were a study of the effect of lubricant coating thickness, using the same AD13 used with specimens 70 through 84. As the thickness of the AD13 was increased from 25 to 127 to 381 nm, the average friction coefficient increased from 0.235 to 0.347 to 0.503. The wear-life data were inconclusive. The similar tests with AD13 over titanium carbide were not directly comparable; the ball-rider materials were different. The average friction coefficients for the 15-g loads on specimens 73, 78, and 83 were 0.238, 0.524, and 0.316 (or 0.524, depending upon the interpretation of "failure"). The TiC-based tests also had erratic and short wear-lives.

The next three groups of three (specimens 61 through 69) were similar to those above, except the lubricant was AD14 for the three thicknesses used. There was no significant difference in the average friction coefficients (0.173, 0.173, and 0.179), with the wear-life data erratic and inconclusive. The similar tests with titanium carbide hard undercoats were on specimens 88, 93, and 98, in which the ball-rider material was 52100 steel and not the titanium carbide used with specimens 61 through 69. The average friction coefficients of 88, 93, and 98 were 0.309, 0.180, and 0.364. The AD14 appeared to give better friction results when applied over tungsten carbide than when applied over titanium carbide. Also, the AD14 appeared to give better friction coefficients than AD13 on both substrates.

In the last 10 groups, various deposition techniques were tried by the supplier. The test conditions were the same for all of the 32 tests and direct comparisons can be made.

Specimens 157 through 161 were coated with alternating layers of gold and a mixture of MoS_2 and Sb_2O_3. The first layer was 7 nm of gold; the second layer was 25 nm of the MoS_2 and Sb_2O_3 mixture. The layers alternated until seven layers of gold and six layers of MoS_2 and Sb_2O_3 had been deposited. Their average friction coefficient was 0.238. The average wear-life was 0.48 min, with a range of 0 to 1.06 min being possible. The use of the gold did not seem to help either the friction or wear-life results.

Specimens 162 through 164 were coated with only the mixture of MoS_2 and Sb_2O_3. The incident angle for the sputtering was 45 degrees. The average friction coefficient was 0.378 and the average wear-life was 0.37 min, with a range of 0.08 to 0.66 min possible.

Specimens 165 through 167 were coated with the same mixture as 162 through 164; the incident angle was increased from 45 to 90 degrees. The average friction coefficient was 0.235; the wear-life average was 0.13 min with a range of 0 to 0.36 min possible. There seemed to be an improvement in the friction coefficient as the path of the material transferred changed from 0 to 45 degrees and on to 90 degrees. However, the wear-life seemed to deteriorate with the increasing incident angle.

Specimens 168 through 170 were coated with the same powder mixture, but the sputtering process was intermittent. Sputtering does impart energy to the part being coated, and the intermittent process seemed to be an attempt to keep the coated surface temperature from increasing. The average friction coefficient cropped to 0.03 for this process, and the wear-life average increased to 5.2 min, with a range of 1.5 to 8.9 min. The "cooling off" periods seemed to help.

Specimens 171 through 173 were coated with the same powder mixture, but in this case, the supplier said the process included "venting." The average friction coefficient was 0.208; the average wear-life was 0.33 min, with a range of times from 0 to 0.73 min being possible. Whatever "venting" meant, it did no good for either friction or wear-life.

Specimens 174 through 176 were coated with MoS_2 and Sb_2O_3 with some teflon powder mixed in. Again, as with the MoS_2 powder question, these are the materials used as the target in the sputtering process; it is not meant to imply that the materials deposited on the surface are the same, either in composition or in crystalline structure. The results indicate that in all probability, neither the MoS_2 nor the teflon was transferred intact. The average friction coefficient was 0.412 for this group. The average wear-life was 0.33 min, with a range from 0 to 0.68 min possible.

Specimens 177 through 179 were coated with MoS_2 and Sb_2O_3 in a "delayed" deposition mode. Again, the results indicate that this technique was not satisfactory. The average friction coefficient was 0.297; the average wear-life was 0.2 min, with a range from 0.1 to 0.3 min.

Specimens 182 through 184 were also "delayed" in their deposition. The atmosphere in which the sputtering was done was an unspecified mixture of argon and oxygen. The results indicate this technique was not satisfactory for these samples. The average friction coefficient was 0.223; the average wear-life was 0.1 min with a range of 0 to 0.2 min.

Specimens 187 through 189 were "delay"-deposited in an atmosphere of argon and BCl_3 (boron trichloride). The average friction coefficient of 0.413 and average wear-life of 2.17 min (ranging from 0.37 to 3.97 min) demonstrated that for these coupons, in this test, the technique was not satisfactory.

The last group of specimens, 192 through 194, was listed as successive deposition and oxidation of MoS_2 and Sb_2O_3. In the preparation of these samples, the process included a little sputtering, a little exposure to air, a little sputtering, etc., until the desired thickness had been deposited. The results were better than some of the other techniques and could conceivably be similar to specimens 171 through 173, the "vented" specimens. The average wear-life was 0.33 min, with a range from 0.27 to 0.39 min. A small range of possible wear-life times, compared to the average time, means that the samples were more consistent in their performance.

C. Conclusions

1. More work is necessary in the use of the sputtering process to improve the quality of the transferred films, the quality required to improve the consistency of both friction and wear, but primarily the consistency of the wear-lives.

2. The latter specimens provided generally gave poorer results than the first specimens provided. The development work seemed to be demonstrating negative progress.

3. The coatings in which titanium carbide was the only material used give longer life and better friction coefficient results than those in which tungsten carbide was the only transferred material.

4. Three instances of better friction and wear-life were demonstrated when rf sputtering was used instead of dc sputtering.

5. Although the lubricating materials to be transferred started as MoS_2, Sb_2O_3, or Teflon, there was no indication in performance, either friction coefficient or wear-life, that these materials were on the coupons used in the tests.

6. Synergistic effects of the mixture of MoS_2 and Sb_2O_3 were apparently demonstrated, with the friction reduced to one-third that of MoS_2 by itself. Work is still needed in the consistency and length of the wear-lives.

7. The different thicknesses used for either AD13 or AD14 had no demonstrated effect as far as friction was concerned. The wear data were inconclusive.

8. The MoS_2 and Sb_2O_3 coatings were better when "thinner" than when "thicker."

9. There was no distinction between the use of high bias voltage and low bias voltage in the sputtering technique.

10. The various technique modifications used by the supplier of the sputtered coatings ("venting," "intermittent," "delayed," etc.) did not improve either friction coefficient or wear-life results.

11. Generally, lower friction results were obtained using 52100 steel ball-riders than when using titanium carbide ball-riders.

SECTION IX

CONCLUSIONS AND COMMENTS

This section summarizes the conclusions that can be drawn from the various projects discussed and presents appropriate comments relative to the work.

A. Conclusions

1. From the Four-Ball repeatability study came the conclusion that there is no statistically significant difference between the two machines used as far as the wear scar data are concerned.

2. The repeatability study revealed that there is a normal amount of data scatter associated with the machines. For the test conditions used as standard, the variation can be $\pm$ 10% of the average wear scar diameter.

3. Based upon the 27 experiments on 20 different greases tested in the Pope spindles, there seems to be more development work required to provide additional greases capable of lubricating a size 204 bearing operating at 10,000 rpm.

4. Wear is the primary cause of the increase in friction in the tapered roller bearing configuration of the Sikorsky testing projects. When wear occurs, the rollers ride back on the cone, contacting the shoulder, where sliding occurs.

5. The Sikorsky test procedure and apparatus are more suited to qualification testing than to research on the operational capacity of greases.

6. Long operating life can be obtained for bearings operating in a vacuum environment without the use of either oils or greases. Two bearings have operated over 54,000 hr each without failure, using lubricant compacts that employ solid lubricants in a powder metallurgy matrix.

B. Comments

1. The three compact-lubricated bearings that stopped operating during this contract period had operating times of 31,075 hr; 32,984 hr; and 52,000 hr. The failures were not due to lubricant breakdown but to the failure of the bonding of the lubricant in the separator.

2. Wear rates (weight loss as a function of time) have been calculated and used to predict the operating lives of the bearings still rotating. The predictions reveal that wear-lives of 13 to 17 years are possible, if the compact inserts do not become disoriented in their separators.

3. If sputtering is to be used as a method for applying thin, uniform coatings on hemispherically-shaped gas bearings, more development work will be required to improve the techniques used on the supplied specimens. Sputtering has been used to produce dry lubricated specimens that have demonstrated successful operation. However, these techniques apparently were not used by the supplier of the gas-bearing coupons used in this work.

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